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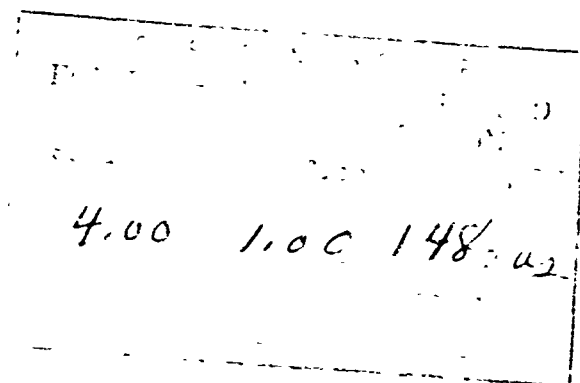
## USAAVLABS TECHNICAL REPORT 65-56

### TRANSMISSION STUDY FOR TANDEM-ROTOR SHAFT-DRIVEN HEAVY-LIFT HELICOPTERS

R-379

By

L. Mack



September 1965

U. S. ARMY AVIATION MATERIEL LABORATORIES

FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-241(T)

VERTOL DIVISION  
THE BOEING COMPANY





DEPARTMENT OF THE ARMY  
U S ARMY AVIATION MATERIEL LABORATORIES  
FORT EUSTIS VIRGINIA 23604

This report represents a part of the USAAVLABS program to investigate mechanical transmission system concepts for a shaft-driven heavy-lift helicopter of the 75,000- to 95,000-pound gross weight class. The purpose of this investigation was to determine the high risk or problem areas that could be expected in the development of a drive train for a mechanically driven heavy-lift helicopter.

This report presents a comparative analysis of various power train configurations for use in a Tandem-Rotor, Shaft-Driven, Heavy-Lift Helicopter. The report also presents discussions of development areas wherein successful accomplishments would render an improved mechanical drive system.

This command concurs with the contractor's conclusions reported herein.

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FORT EUSTIS, VIRGINIA

## SUMMARY

This report presents the results of a study of mechanical drive systems for a tandem-rotor, heavy-lift helicopter (HLH). This work was performed for the U. S. Army Aviation Materiel Laboratories (USAAVLABS) by the Vertol Division of Boeing, under Contract DA 44-177-AMC-241(T), between 27 June 1964 and 27 December 1964.

Design characteristics for the helicopter and description of the mission were given by USAAVLABS. Drive system configurations were selected by the contractor.

The objectives of the study were as follows:

1. To define problem areas peculiar to a helicopter capable of fulfilling the mission requirements
2. To indicate development programs necessary to eliminate high-risk problem areas
3. To provide sufficient weights information to permit comparison of mechanical drive systems with other drive systems

The primary conclusions drawn from this investigation are as follows:

1. The confidence level is high for the satisfactory solution of the tandem heavy-lift-helicopter mechanical drive system. A comparison has been made of the HLH and current technology, using certain factors which generally indicate state of the art. The HLH does not exceed present levels in the more significant factors.
2. Programs of drive system component development are required in three areas to ensure high confidence. These are: gear surface durability, engine control, and overrunning clutches. Other investigative areas are defined in this study as highly recommended, but not essential to the HLH.
3. Confidence in the tandem-rotor drive system is increased because total power is shared between two

rotor transmissions. This minimizes the required extension to present day drive system technology.

4. Specific weight of the study drive system approximates 0.58 pound per horsepower.
5. Advanced gear, bearing, and shafting technologies contribute to a 10-percent reduction in drive system weight.
6. Supercritical speed shafting provides a 160-pound weight decrease compared to subcritical designs. A concurrent improvement in system reliability is expected by reduction in dynamic components.
7. Final reduction ratios required can be accommodated by conventional gearing arrangements. However, high-ratio devices competitive in efficiency to present standards are particularly applicable to the HLH.
8. Bevel gearing capable of transmitting HLH power does not exceed current experience levels, except that surface durability (scoring) requires investigation and possible corrective action.

Investigations of (1) gear tooth surface capacity, (2) multi-engine control, and (3) overrun clutches are recommended to achieve confidence in the reliability of the HLH drive system:

1. Gear tooth scoring is an important surface failure phenomenon. The requirements of the HLH indicate main-power gearing which is within the presently accepted scoring region. To avoid this problem, and maintain minimum-weight transmissions, it is necessary to improve scoring resistance by analysis and test of promising solutions.
2. A requirement for more than two engines will increase pilot responsibility for power management and matching. To determine the magnitude of the problem, it is recommended that a multiengine simulator be used. Further, if this problem proves serious, automatic control methods must be investigated for the engines applicable to the HLH requirement.

3. Increased torque requirements and placement of the clutch at the transmission input combine to raise rubbing velocity over the level of present experience. Clutch life and reliability must be evaluated at contemplated overrun velocities. Clutch positioning at the transmission-engine interface presents dynamic unknowns which also require study.

In addition, the study indicates that investigations of (1) gear tooth strength, (2) supercritical-speed shafting, (3) bearing analysis, and (4) materials improvement are particularly worthy of effort, and that they will provide maximum return to HLH and other drive systems:

1. To assist in obtaining the required improvement in surface durability and to obtain higher load-carrying capacity for the same gearing weight, a program to improve gear tooth bending fatigue strength is recommended.
2. The power requirements and length of the HLH combine to increase the significance of interconnect shafting. Supercritical speed operation, by eliminating shaft supports, reduces weight. By reducing potential failure points, reliability is increased. Continuation of present efforts through a flight demonstration is recommended.
3. Larger bearings operating, in some cases, at higher speeds require rigorous analytical approaches to ensure reliability. Confident use of available increases in capacity requires more complete knowledge of loads and velocities within the bearing.
4. The use of titanium is being widely explored for drive system application. Successful application in many areas which can take advantage of its unique characteristics will provide substantial weight reduction to the drive system. Materials and processes in the area of ferrous metals promise greater strength-weight ratios, increased predictability, and dimensional stability in large sizes. Continuing programs to evaluate promising approaches should be encouraged. The results of these programs can be expected to benefit current helicopter drive systems,

as well as the heavy-lift helicopter of the future.

This study has investigated several tandem-rotor HLH concepts. A high level of confidence has been demonstrated for the successful application of the mechanical drive systems investigated and in all areas the design solutions are within current technology. However, the drive systems are also subject to improvement. Programs to advance technology are suggested.

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## INTRODUCTION

The helicopter and rotor system on which this transmission study was based is, in part, the result of the mission requirements and general characteristics supplied by USAAVLABS and, in part, the result of past Vertol Division study. A continuing program of study may modify certain characteristics of the heavy-lift helicopter (HLH), as study conclusions and recommendations are assimilated. It is therefore desirable to indicate the sensitivity of the drive system study conclusions to such design characteristics. Rotor disc loading and rotor thrust are selected for certain conditions. If the conditions change, what will be the effect on the drive system?

Figure 1 illustrates a comparison of the HLH and current technology. Major study configurations I, IA, II, and III are shown in Figures 2, 3, and 4.

Drive system torque is effectively independent of rotor disc loading for a constant gross weight and rotor tip speed. While the power required increases with smaller diameters (higher disc load), the rotor rpm increases to maintain a constant tip speed. The relationship between induced rotor torque, thrust (gross weight), rotor radius, and rotor speed is shown in equation (1). Total torque increases slightly with disc load, because total power includes profile power.

$$\text{Torque} = \frac{\frac{\text{Thrust}}{2}}{(2\rho\pi)^{\frac{1}{2}}(\Omega R)} \quad (1)$$

The torque-power relationship is:

$$\text{Torque} = \frac{P_i}{\Omega} \quad (2)$$

Rotor power requirement is based upon:

$$P_i = (\text{Thrust})V_i \quad (3)$$

$$V_i = \sqrt{\frac{\text{Thrust}}{2\rho\pi R^2}} \quad (4)$$

where

$P_i$  = induced power

$\rho$  = density of air

$V_i$  = induced velocity

$R$  = rotor radius

$\Omega$  = radians/second

Study areas (such as gear sizes) which depend upon torque are not greatly influenced by variations of possibly  $\pm 20$  percent in rotor disc loading, at constant thrust. This indicates that rotor diameter alone is not a powerful factor in the study conclusions.

A change in thrust (or gross weight) affects the torque requirement, assuming constant rotor disc loading, constant tip speed, and aerodynamically equivalent rotors. A 20-percent increase in mission weight would, by reference to the foregoing equation, indicate that torque is increased by 32 percent. The effect of aircraft gross weight change on drive system conclusions is therefore more significant than is the effect of rotor disc load. While considering this, it is well to remember that gear diameter is responsive to the cube root of torque (Figure 5). Therefore, a one-third increase in torque will increase gear diameter and velocity by 10 percent. Since this is exactly the amount rotor rpm decreased in the example, conclusions relating to gear velocity are still relevant.

The effect of rotor diameter influences the transmission system arrangement in a less obvious manner. In the early study configurations it became apparent that system simplifications could be made by directing the engine inputs to the rotor gearbox, thereby eliminating engine nose boxes and bevel gearing. To do this it was necessary to house the engines in an airfoil section protruding from the fuselage. The effect of rotor downwash on this section is dependent upon rotor diameter. The rotor diameter used in the study vehicles is large enough to ensure that downwash is not significant. With a lesser diameter, it would become increasingly significant. Therefore, the final transmission arrangement would be the result of trade-off studies between rotor diameter, drag area, fuselage length, and transmission complexity.

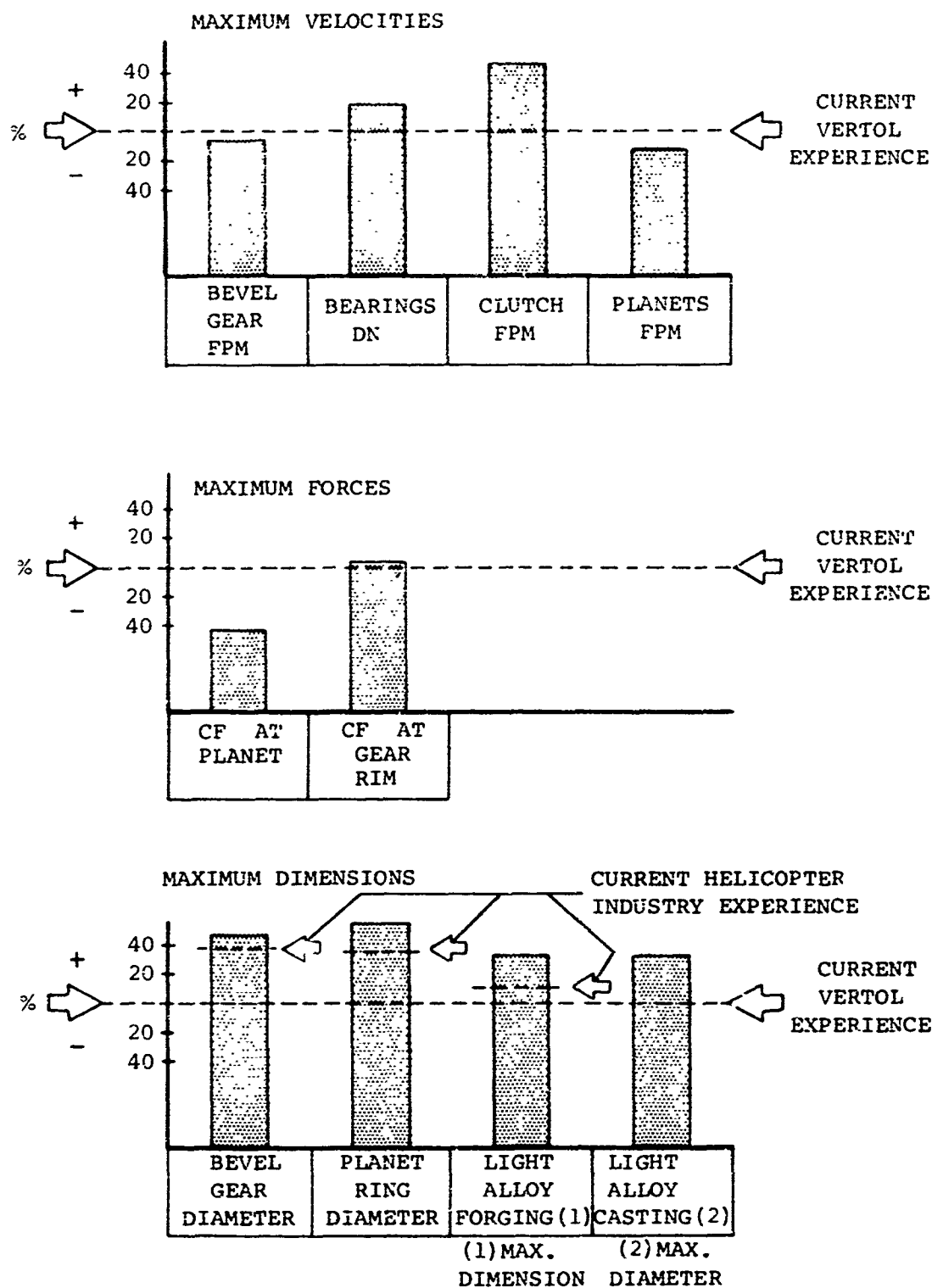


FIGURE 1. COMPARISON OF HLH DRIVE SYSTEM WITH CURRENT SYSTEM.

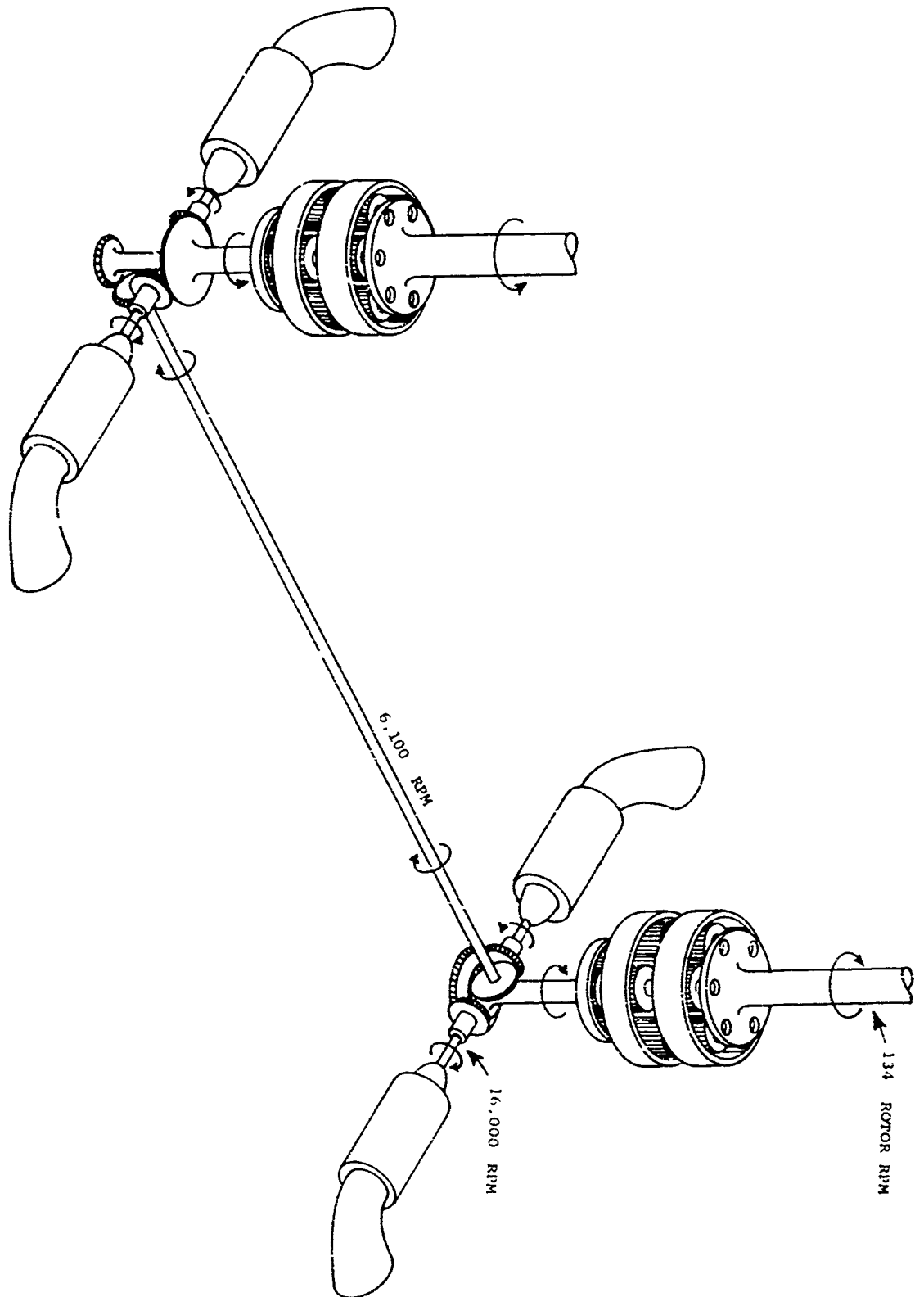


FIGURE 3. STUDY CONFIGURATION II.

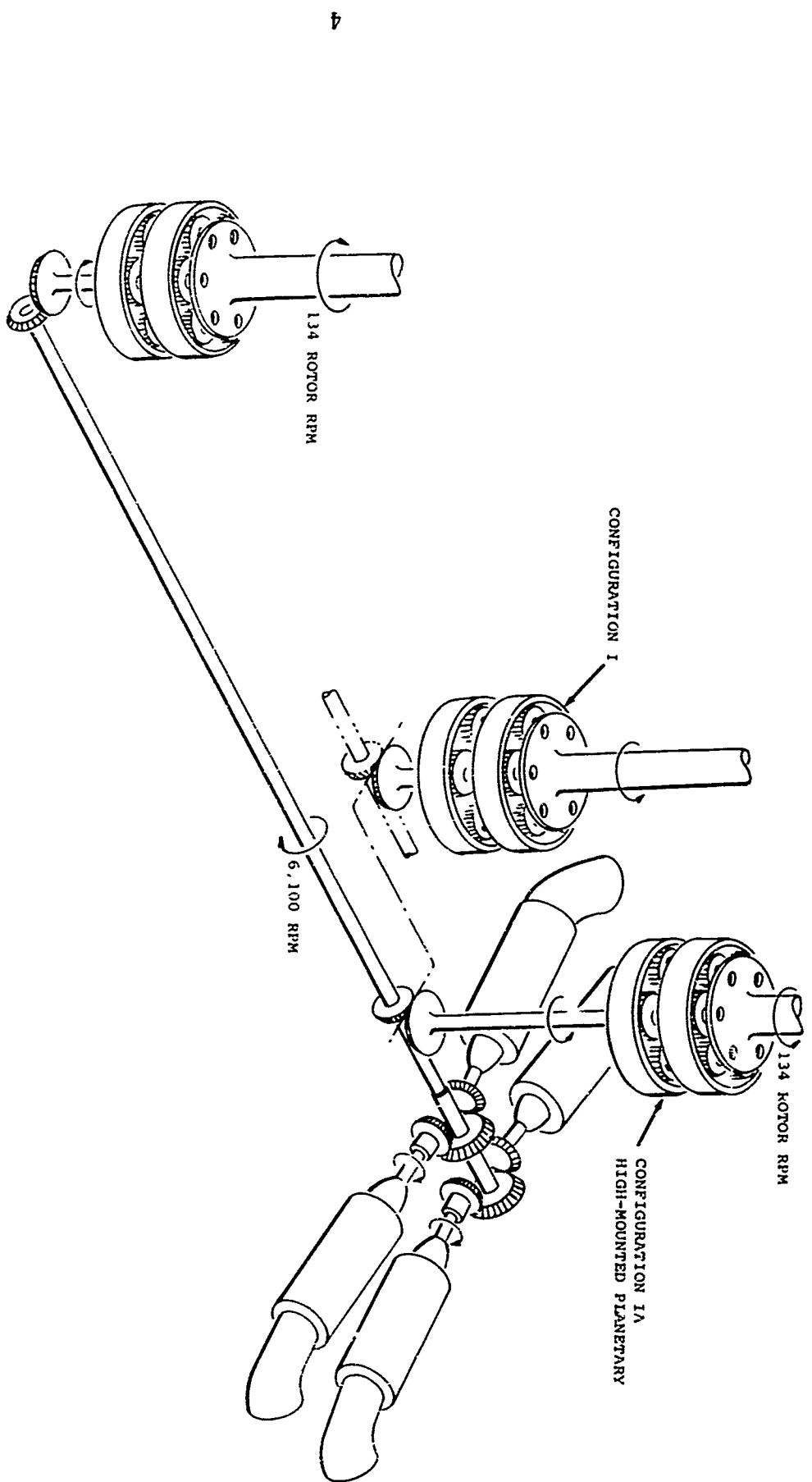


FIGURE 2. STUDY CONFIGURATIONS I AND IA.

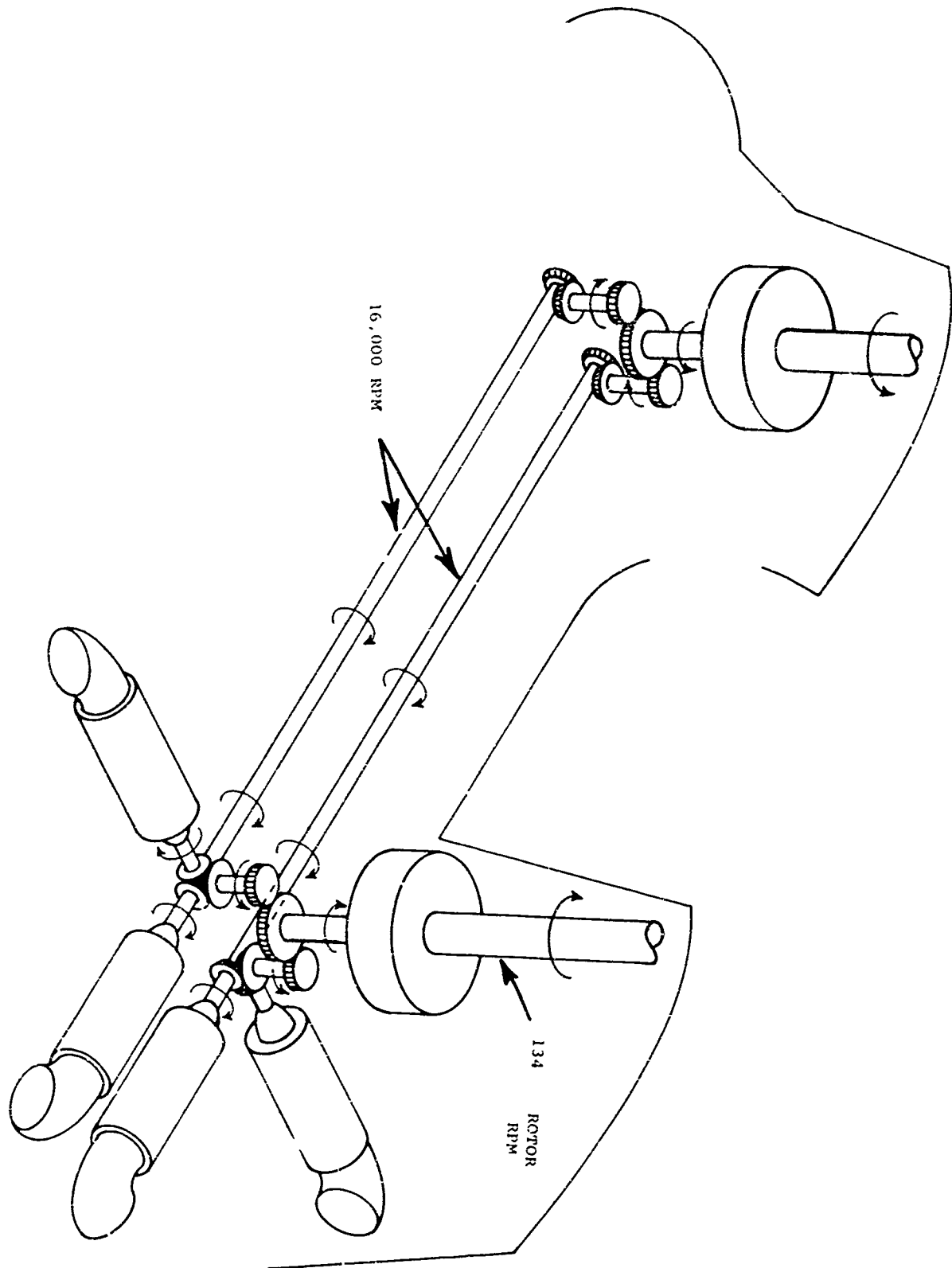


FIGURE 4. STUDY CONFIGURATION III.

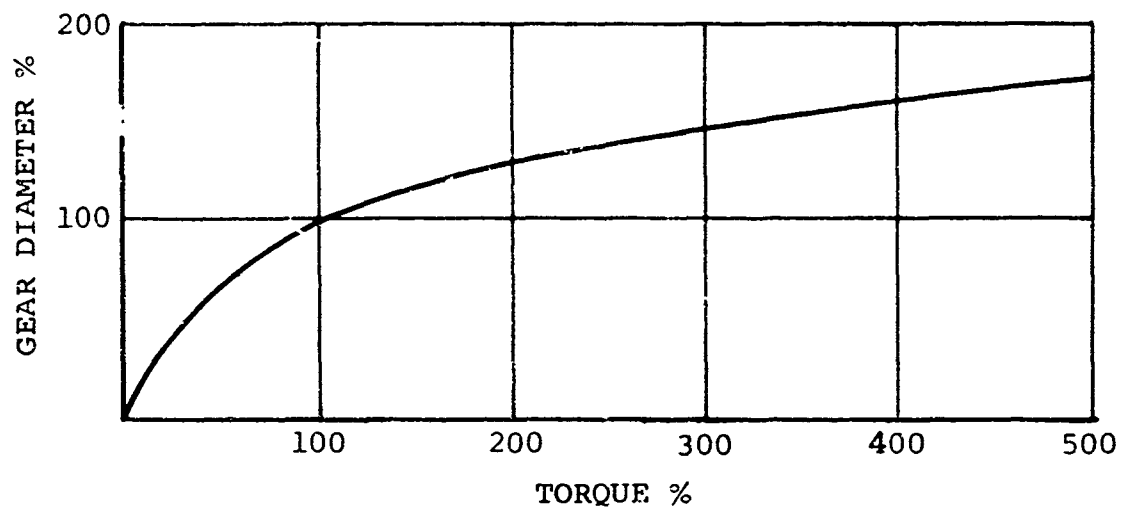


FIGURE 5. EFFECT OF TORQUE ON GEAR DIAMETER (FOR A CONSTANT FACE/DIAMETER RATIO).

### DATA SUPPLIED

The following helicopter design characteristics and mission description were furnished by the study contract document:

#### DESIGN CHARACTERISTICS

1. Gross weight: 75,000 to 85,000 pounds
2. Turbine-powered
3. Safe autorotation at design gross weight
4. Design load factor: 2.5 at design gross weight
5. Crew minimum of 1 pilot, 1 copilot, and 1 crew chief
6. All components to be designed for 1200 hours between major overhaul, and 3600 hours' service life

#### MISSIONS

##### Transport Mission (Figure 6)

1. Payload: 12 tons (outbound only)
2. Radius: 100 nautical miles
3.  $V_{cruise}$ , 12-ton payload: 110 knots
4.  $V_{cruise}$ , no payload: 130 knots
5. Hovering time: 3 minutes at takeoff, 2 minutes at midpoint (with payload)
6. Reserve fuel: 10 percent of initial fuel
7. Hover capability: 6000 feet, 95°F. (OGE)
8. Mission altitude: sea-level standard atmosphere
9. Fuel allowance for start, warm-up, and takeoff: MIL-C-5011A

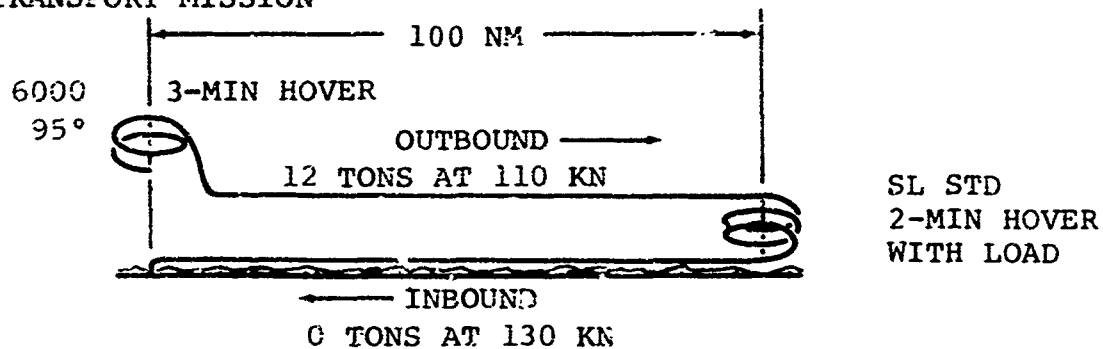
##### Heavy-Lift Mission (Figure 6)

1. Payload: 20 tons (outbound only)
2. Radius: 20 nautical miles
3.  $V_{cruise}$ , 20-ton payload: 95 knots
4.  $V_{cruise}$ , no payload: 95 knots
5. Hovering time: 5 minutes at takeoff, 10 minutes at destination (with payload)
6. Reserve fuel: 10 percent of initial fuel
7. Hover capability: sea level, 59°F. (OGE)
8. Mission altitude: sea-level standard atmosphere
9. Fuel allowance for start, warm-up, and takeoff: MIL-C-5011A

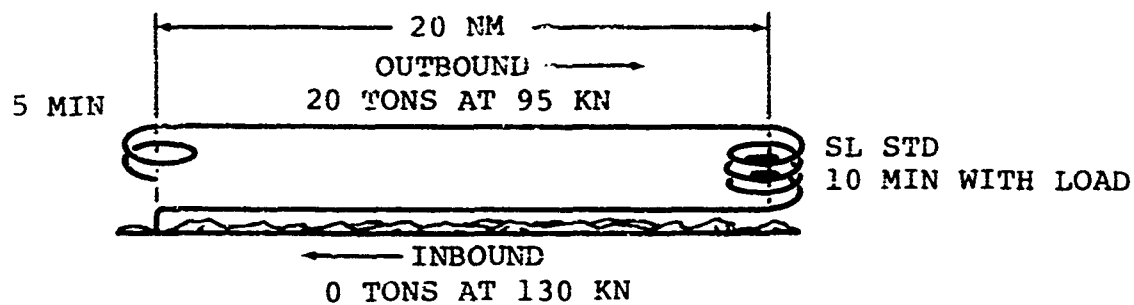
### Ferry Mission (Figure 6)

1. Ferry range (no payload, STOL takeoff):  
1500 nautical miles
2. Reserve fuel: 10 percent of initial fuel
3. Fuel allowance for start, warm-up, and takeoff:  
MIL-C-5011A
4. Minimum design load factor of 2.0
5. Mission altitude: sea-level standard atmosphere
6. Best speed for range

#### TRANSPORT MISSION



#### HEAVY-LIFT MISSION



#### FERRY MISSION

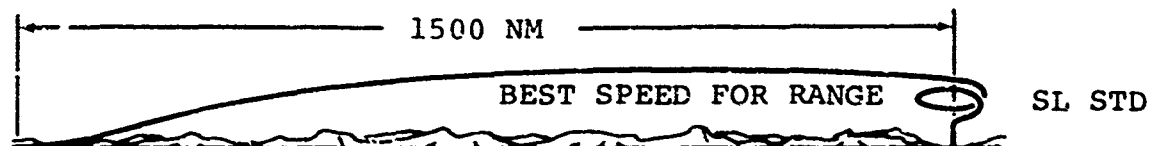


FIGURE 6. SPECIFICATION MISSIONS.

## ANALYSIS OF PROBLEM

### REQUIREMENTS OF HLH

The transmission study configurations are based upon the following requirements:

1. Maximum design horsepower: 15,200
2. Rotor configuration: Tandem
3. Rotor diameter: 96 feet
4. Rotor rpm: 134

The maximum design horsepower (15,200) is based upon the power available at standard conditions from advanced versions of current engines. The installed horsepower requirement is derived from the specified 6000-foot, 95-degree, transport hover. The maximum sea-level horsepower required by the mission characteristics is 11,500. The rotor system will be designed to have an eventual capability of absorbing the available 15,200 shaft horsepower (shp). This additional power can be utilized to increase aircraft capability by providing higher speed or greater payload-carrying capacity than specified in the mission definition. Designing the drive system to 15,200 shp provides assurance that the study conclusions reflect the most arduous condition expected. System weights must be factored for comparison with systems designed to lesser power requirements.

Rotor diameter was based upon a conservative disc loading for the gross weight supplied. This loading, 5.55 pounds per square foot at 80,000 pounds' gross weight, is directly comparable to Vertol Division experience for the CH-47A Chinook. As in the Chinook, it allows margin for efficient growth versions at higher gross weights. Rotor rpm is also determined from Vertol Division experience, and results in a tip speed of 700 feet per second.

### OBJECTIVES

A necessary preliminary to the design of the system is the establishment of the design objectives. These are used as a guide during the conceptual study phases, and as a check list for evaluation in the final design stage. The overall objectives for the HLH drive system are that it shall perform

its functions with maximum reliability, minimum total effective weight, and minimum maintenance.

The functions of the drive system are to:

1. Distribute power from prime movers to rotors
2. Make the necessary speed reduction
3. Synchronize rotors
4. React rotor forces to airframe
5. Slow and stop rotors
6. Power aircraft accessories

#### System

The means by which objectives will be attained in the system are as follows:

1. Reduce gear mesh loss to a minimum by arrangement of the system and choice of intermediate ratios.
2. Reduce number of gearcase assemblies to a minimum.
3. Increase reliability by approaches such as inspection systems for incipient failure, use of material with inherently slow crack propagation, and redundancy.
4. Maintain system rotational speeds at a maximum up to the final reduction, consistent with final drive ratio available.
5. Eliminate power-mixing functions on interconnect shaft.

#### Subsystem

The means by which objectives will be attained in the subsystems are as follows:

##### Transmission Assemblies

1. Use multiple load paths wherever possible. The failure of any single component shall not be catastrophic in effect.
2. In addition to, or in lieu of, the preceding paragraph, ensure that the fatigue failure propagation rate of the component shall be slower than the normal inspection period.

3. Provide failure warning of nonvisible components.
4. Increase material reliability by advanced processing techniques.
5. Eliminate or damp resonant frequencies within the operating range.
6. Consider emergency nonlubricated operation, especially for bearings.
7. Increase power/weight ratio by equalized loading of planet gears.
8. Develop high-ratio, high-efficiency final reduction devices.
9. Minimize weight of rotor support and reaction structure.
10. Consider the following maintenance requirements: (1) increased TBO - 1200 hours between overhauls, 3600 hours to retirement, and capability of reliable operation between 300-hour periods; (2) field equipment and skill limitation; and (3) accessibility of components.

#### Shafting

1. Utilize supercritical speed operation for less weight and increased reliability. However, all shaft systems shall also have provision for subcritical operation by addition of supporting members.
2. Minimize number of shaft joints and flexible couplings, consistent with deflection and handling requirements.
3. Consider the effect of end supports when critical speed is calculated, especially for high-speed application.
4. Incorporate redundant design wherever practicable, especially in the attachment areas.
5. Make shaft couplings and attachments visually inspectable.
6. Increase misalignment capacity in shaft couplings and spline joints.
7. Improve lubrication of splined joints where misalignment is to be accommodated.
8. Reduce balance requirements of shafting by supercritical speed operation.

9. Reduce or eliminate lubrication maintenance of shaft assemblies.

#### Lubrication System

1. Jets shall be multiple or shall be removable and inspectable from the outside of the gearcase.
2. Filters shall have indication of condition, preferably in cockpit.
3. Oil temperatures shall not exceed the present maximum.
4. Internal lubrication passages shall be machined and inspectable.
5. Situation of the oil cooler and blower shall be as close as possible to each transmission.

#### Rotor Brake

Rotor brake design objectives shall include the following:

1. The rotor brake shall not be mounted on a main drive shaft.
2. Brake actuation and control shall be simplified and failsafe.
3. Consider shielding of brake disc.

#### MECHANICAL DESIGN CRITERIA

Design criteria for the HLH drive system was obtained from a power-required analysis for each phase of the flight regime. Gears, splines, and shafts were designed for unlimited fatigue life, and 3600-hour wear life. Bearings were designed according to a cubic mean power taking into account all expected powers shown in Tables I and II.

The cubic mean power is derived from Palmgren (Reference 5) and from others as follows:

$$F_m = \sqrt[3]{\frac{F_1^3 N_1 + F_2^3 N_2 + \text{-----}}{N}} \quad (5)$$

where

$F_m$  = cubic mean power  
 $F_1$  = power acting  $N_1$  % of time  
 $F_2$  = power acting  $N_2$  % of time  
 $N$  = 100% of time

The minimum bearing design life was 1200-hour B10 (B10 is the life that 90 percent of the bearings of an apparently identical lot can be expected to meet or exceed). This design life is consistent with the contractual requirements and current design objectives. The design criteria "Percent of Time" was derived from analysis of the expected overall mission profile of the HLH. This analysis was used to estimate the amount of time which the HLH would spend upon each general task during its total service life. For example, percent of time performing the heavy-lift mission, percent performing the transport mission, and percent in ferry to and from the theatre of operation. It will be noted that a high percentage (31 percent) of the total life was estimated for the heavy lift hover which requires 11,500 shp. This decision indicates the tendency to provide a conservative bearing design criteria.

The maximum available sea-level power of 15,200 shp is predicted for only 5 percent of the time. Its effect on the cubic mean power is arithmetically negligible. Bearings associated with gear support functions were not, therefore, affected by the maximum power criteria. Rotor shafts and rotor shaft support bearings were sized by preliminary aerodynamic analysis of rotor loads, as well as by the maximum torque requirement. A rotor shaft bearing load schedule, consistent with the expected flight conditions, was set up and used to obtain required bearing capacities. The power distribution between rotors was determined after a review of flight test data from current tandem rotor helicopters and after consideration of any special requirements of the HLH. A distribution factor consistent with present experience was decided upon.

Gears, shafts and splines are designed for continuous operation at that proportion of maximum takeoff power (15,200 shp) dictated by their position in the system and by the application of the distribution factor (in the rotor transmissions).

TABLE I. MISSION DESIGN CRITERIA

Flight Condition	Percent of Time	Rotor Tip Speed (fps)	SHP for Std Day SL
Climb	5	700	11,500
V <sub>max</sub>	5	700	15,200
V <sub>cruise</sub>	40	700	7,100
Hover			
Transport mission	14	700	9,800
HL mission	31	700	11,500
Autorotation	5	875	0

Remarks:

1. The power distribution to either rotor is to be based upon current data.
2. Alternating torque is to be considered as 15% of mean for fatigue life.
3. Rotor shaft included angles (measured from aircraft waterline) to be:  
     forward: 100-1/2°  
     aft: 85-1/2°
4. Rotor rotation shall be determined by the particular transmission configuration.

TABLE II. BEARING DESIGN CRITERIA

Condition	Cubic Mean Power	Tip Speed	Life B10
Normal	10,000	709	1,200
Engine(s) out	Single-engine power	700	300

They are not affected by cubic mean load schedules.

#### DESIGN STRESS LEVELS

A safe stress level is based upon experience with a particular type of design and with a particular method of calculation. It is considered safe if final performance meets standards of reliability and maintenance-free operation predicated as satisfactory for the product. Since considerations of reliability and maintainability are paramount in aircraft, the strong tendencies for the designer are to formulate conservative stress levels and use conservative analysis techniques. This is especially true because of the immediate pressures upon the aircraft manufacturer to produce a safe and economically feasible mechanism with a minimum of time available for development and uprating of component stress levels. However, in the course of programs such as those proposed in the present HLH study, an opportunity will be presented to evaluate improvements in materials and technology realized in the past decade and to further develop proven concepts in order to yield lower-weight, more reliable mechanical drive systems.

To this end the Vertol Division has used the results of its own and others' experience to propose attainable goals in stress levels for the time period of the HLH. This is understood to be the 1970 era, or approximately 8 to 10 years beyond the basic design of current medium-capacity helicopters. The goals established have been used in the design studies of this report, and have been compared to 1960-era levels in terms of weight, and of other significant criteria (Figure 7).

These goals are predicated upon the following:

1. Developments of the past few years have indicated that the improvement potential exists.
2. No technical breakthroughs will be required for their attainment; rather, an evolutionary-type improvement will be required to substantiate these present indications.
3. An aggressive development program will be supported to realize this evolutionary improvement within the

time permitted. To implement this program, test and development facilities in existence, and planned, should be utilized to provide a level of confidence which will permit advanced stress levels to be applied to the HLH.

It should be emphasized that the full use of advanced stress levels represents weight and size advantages to the HLH. These stress levels are not, in the main, prerequisites to the successful solution of the HLH drive system; a solution making use of 1960-era stress levels is practicable. This is illustrated by Figure 35, which delineates an HLH rotor transmission designed to current stress levels. This may be compared to Figure 36 which shows the same transmission using advanced levels. Figure 8 compares the weights of these two transmissions. The reduction in planetary ring gear size from 38 inches to 33 inches by a combination of improvements in bearings and gears is especially noteworthy. The effect of advanced stress levels on the spiral bevel gears is shown in Figure 9.

Confidence in the feasibility of obtaining more from a material presupposes increased knowledge of the material, and of the environment in which it operates. If this confidence is borne out by testing, the increased capability can be used to provide greater reliability at a stress level comparable to that now used. Two options can therefore be derived from a successful development program: improved power to weight, or improved reliability. The ultimate trade-off between these two will be influenced by field experience accumulated before the final design decisions are made, as well as by the success of the development effort in meeting and exceeding the objectives.

Areas where increased stress levels were investigated are the following:

1. Spiral bevel gearing
2. Spur and helical involute-form gearing
3. Rolling-element bearings

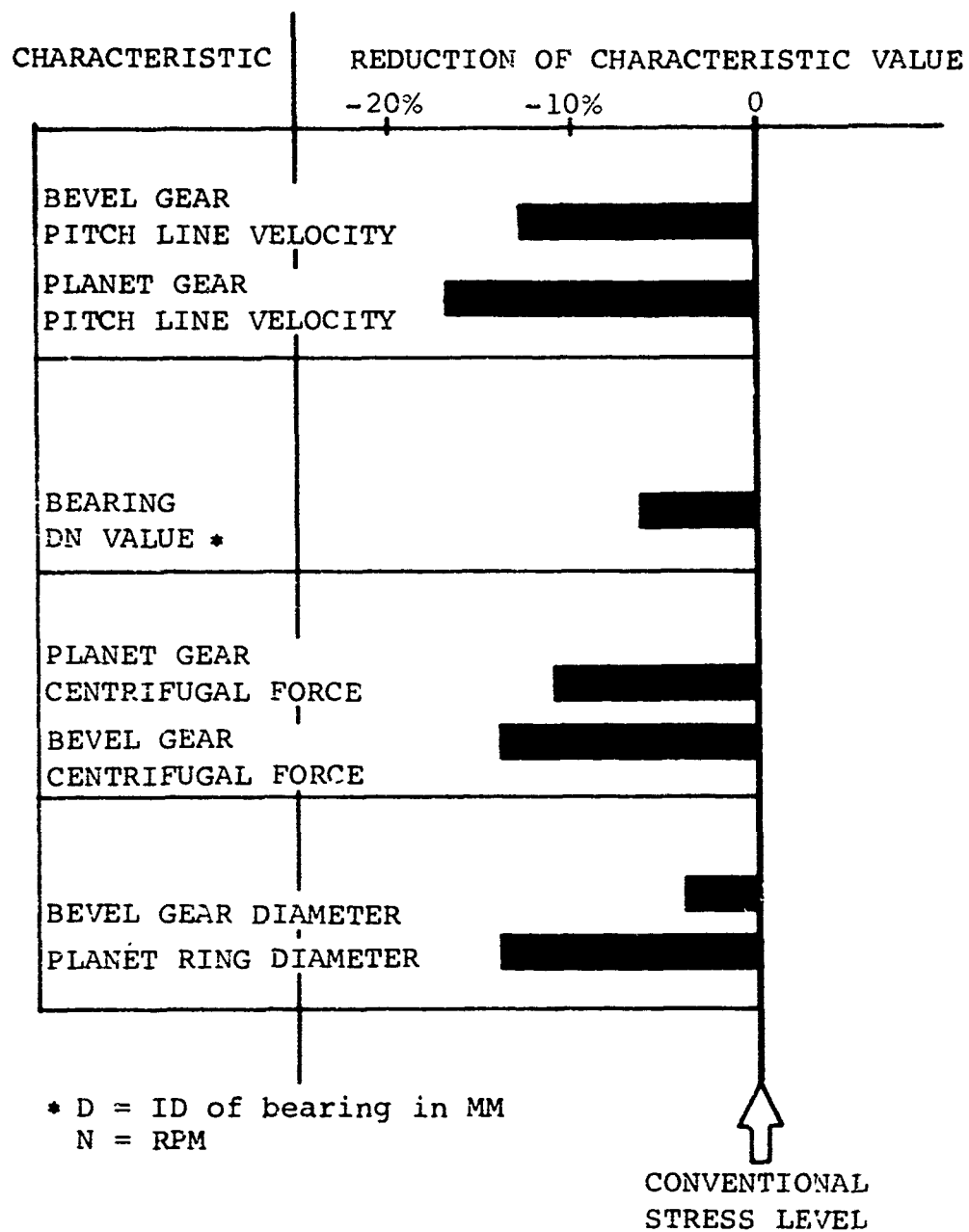


FIGURE 7. ADVANCED STRESS LEVELS - EFFECT ON DRIVE SYSTEM CHARACTERISTICS.

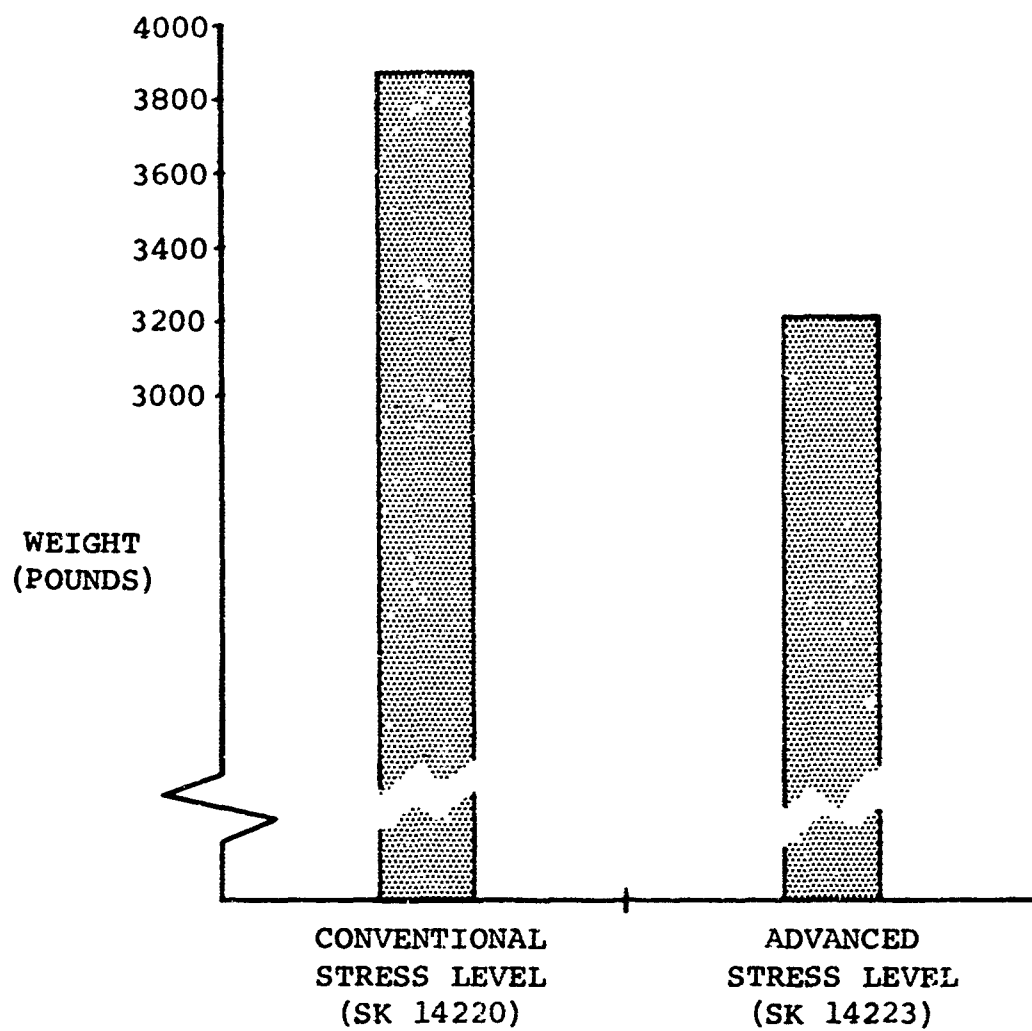


FIGURE 8. WEIGHT EFFECT OF STRESS LEVELS - FORWARD TRANSMISSION.

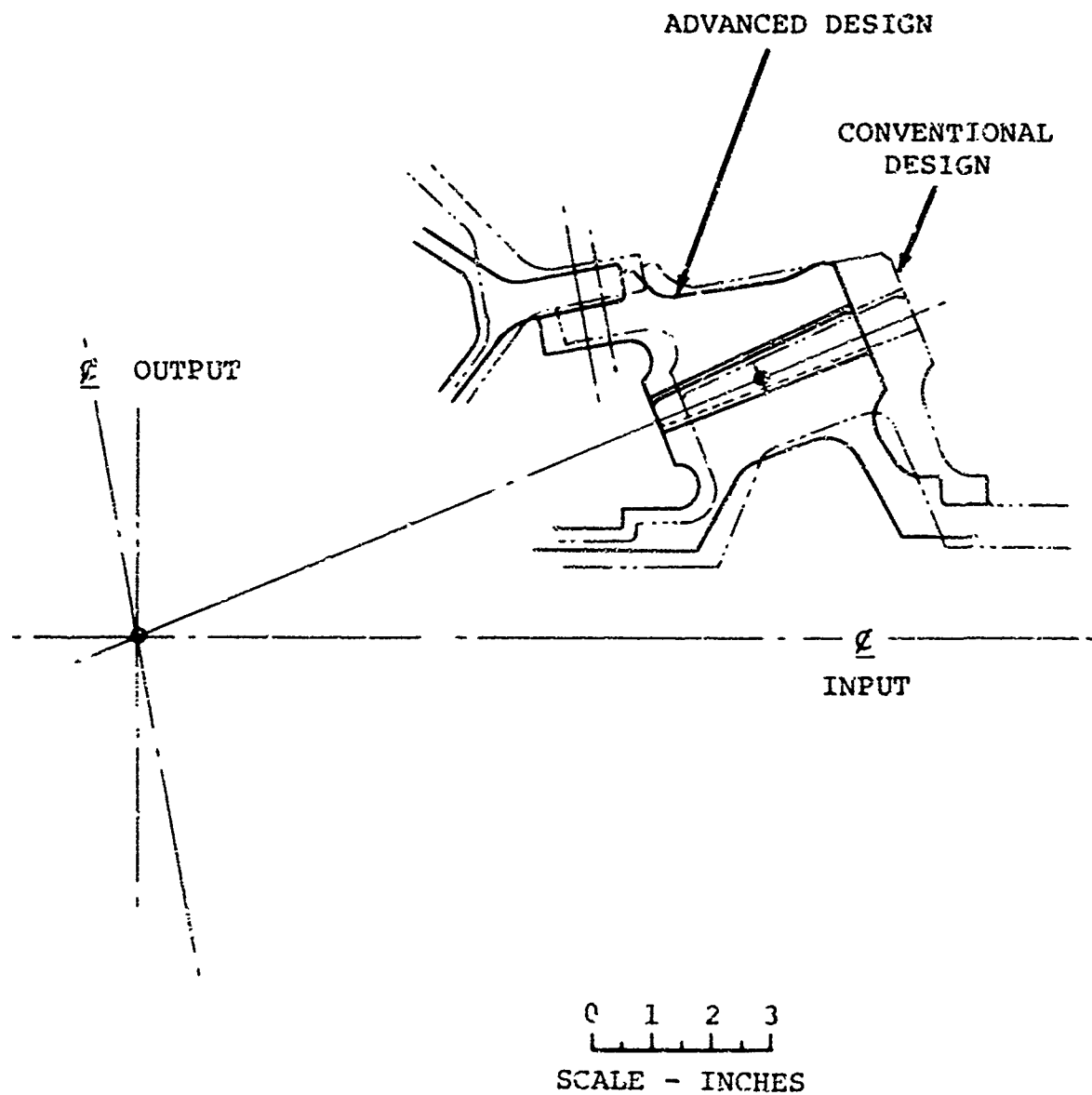


FIGURE 9. COMPARISON OF BEVEL GEAR SIZES.

Areas where increased stress levels were not used are:

1. Shafts under combined loading with possible fretting at connections, and with overriding rigidity and spring rate design criteria.
2. Gear carriers and webs under combined loading.
3. Light alloy forgings and castings.

It is believed that successful application of new material combinations and techniques to the first group could influence the acceptable limits placed upon the second. However, there is generally in the second group a less direct and less provable (particularly by component testing) relationship between stress level and service reliability. There is also a possible degradation of fatigue endurance by reason of the size factor in the large forgings and castings required for the HLH drive system. It was, therefore, assumed for this study that such elements would be developed and extended during future growth of an HLH vehicle in being.

Information on the progress being made in development of advanced stress levels was obtained from the following sources: major bearing manufacturers, the Gleason Gear Works (Reference 8), Vertol Division's experience in growth design and in development testing. In consulting the outside sources, the following questions were asked: How does your present day experience compare with published design allowables (or lives); what is your estimate for growth in the next 5 years; upon what technical advancements is your growth prediction based; what does existing test data indicate for growth potential?

#### Bearings

On the question of growth in bearing capacity, the consensus is that a life expectancy of 3 to 10 times present catalog ratings could be realized by 1970. New materials and techniques have shown increases of 20 to 30 times rated life, in test lots. Life improvements have been twofold or threefold in the past few years, but these have been hidden by more difficult operating conditions, such as the use of MIL-L-7808 oil, and at times, by unforeseen dynamic loads and misalignments. Qualified in terms of load for the same design life,

future bearings may be expected to carry 1.4 to 2.2 times present loading.

Bearing life-load relationship is derived from the following equations (Reference 5):

1. For ball bearings:  $L = \left(\frac{C}{P}\right)^3$  (6)

2. For roller bearings:  $L = \left(\frac{C}{P}\right)^{3.33}$  (7)

where

L = life  
C = load rating  
P = load

The methods of attaining bearing life and load improvement will include the following:

1. Reduction of unwanted inclusions by melting steel in vacuum.
2. The use of new types of bearing steels such as M-50.
3. The systematic improvement of existing (52100) bearing steel by evaluation of the effect on fatigue life of trace elements, and by modification of the major chemical constituents (Reference 3). Trace elements have been present in the chemical analysis because past methods of steel making could not eliminate them. New methods, including vacuum melting, are now able to control or exclude them.
4. Elimination of end grain and segregated material, by new forging techniques for raceways. This is particularly pertinent to ball thrust bearings, wherein the ball elements run against the side of the race, and where end grain exists when races are machined from tube.
5. Manufacturing techniques to improve bearing element finish and roundness.
6. Peening of bearing surfaces to improve residual stress conditions and possibly disperse carbides tending to form fatigue nuclei, and to improve end grain effects.

These, and other approaches, typify the manufacturing techniques which have already indicated significant bearing life-load capacity improvement. Some are presently in use in the Boeing-Vertol CH-47A. Figure 10 illustrates the degree of improvement expected from various approaches by one bearing manufacturer (Reference 5).

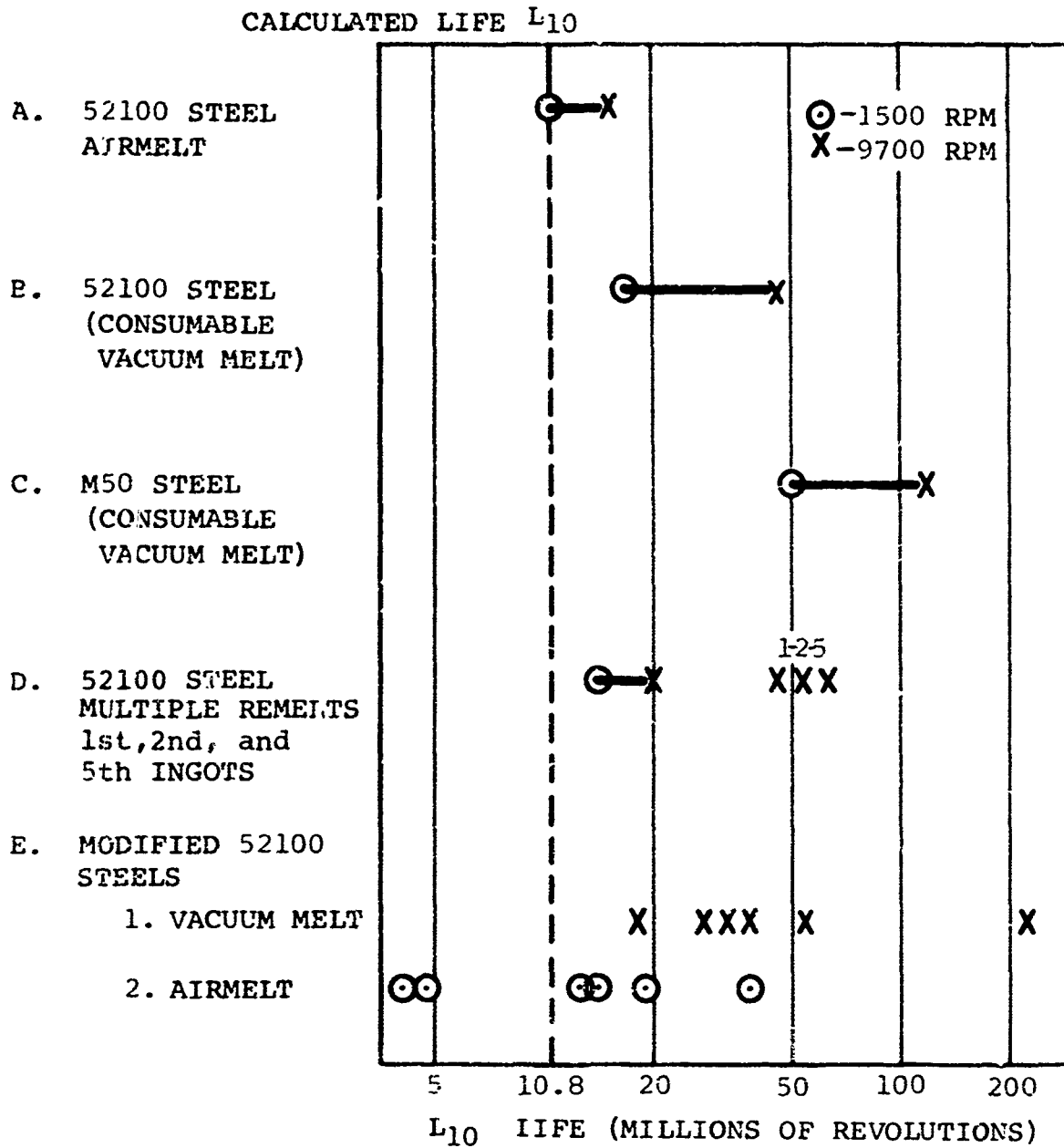
The question of bearing predictability, or the spread in test life experienced in an apparently identical bearing lot was discussed. The majority opinion was that little decrease in this spread has been noted, despite the life increases experienced as a result of material and design improvement. A significant improvement in predictability apparently awaits a basic explanation of bearing fatigue causality; this has yet to be made.

Apart from the material and processing improvements described, analytical techniques must keep pace to ensure utilization of bearings to maximum advantage. The theory is in large part known, and requires systematizing and programming to make it available for design use. With existing and future programs, the external forces upon the bearing, and internal forces velocities, and directions of the rolling elements must be quantified with exactitude. The method of analysis will include the rigidity of the races, centrifugal and gyroscopic forces upon the elements, and other effects which differentiate a real bearing from the simplifying assumptions of normal calculation methods. This difference becomes of major importance with high-speed, highly loaded bearings operating with combined loading and with varying degrees of housing rigidity.

#### Spur and Helical Gearing

The power-carrying capacity of a given gear depends upon gear tooth strength, and gear tooth surface contact capacity. Testing conducted by Vertol and others has given strong indication that aircraft gear tooth bending strength is higher than generally assumed. For example, testing recently concluded under Contract DA-23-204-AMC-02693(T) by Vertol Division has shown a mean failure level approximately five times higher than normal design levels (Figure 11). This program was designed to evaluate various gear tooth grinding effects in comparative testing of basically similar gears.

(INFORMATION COURTESY OF SKF INDUSTRIES,  
INC., KING OF PRUSSIA, PA.)



POINTS REPRESENT ESTIMATED LIFE OF INNER RINGS FROM SAME HEAT OF STEEL (MINIMUM LOT TESTED WAS 30 BEARINGS), OR THE AVERAGE OF SEVERAL LOTS.

FIGURE 10. BEARING LIFE IMPROVEMENT BY IMPROVED METHODS.

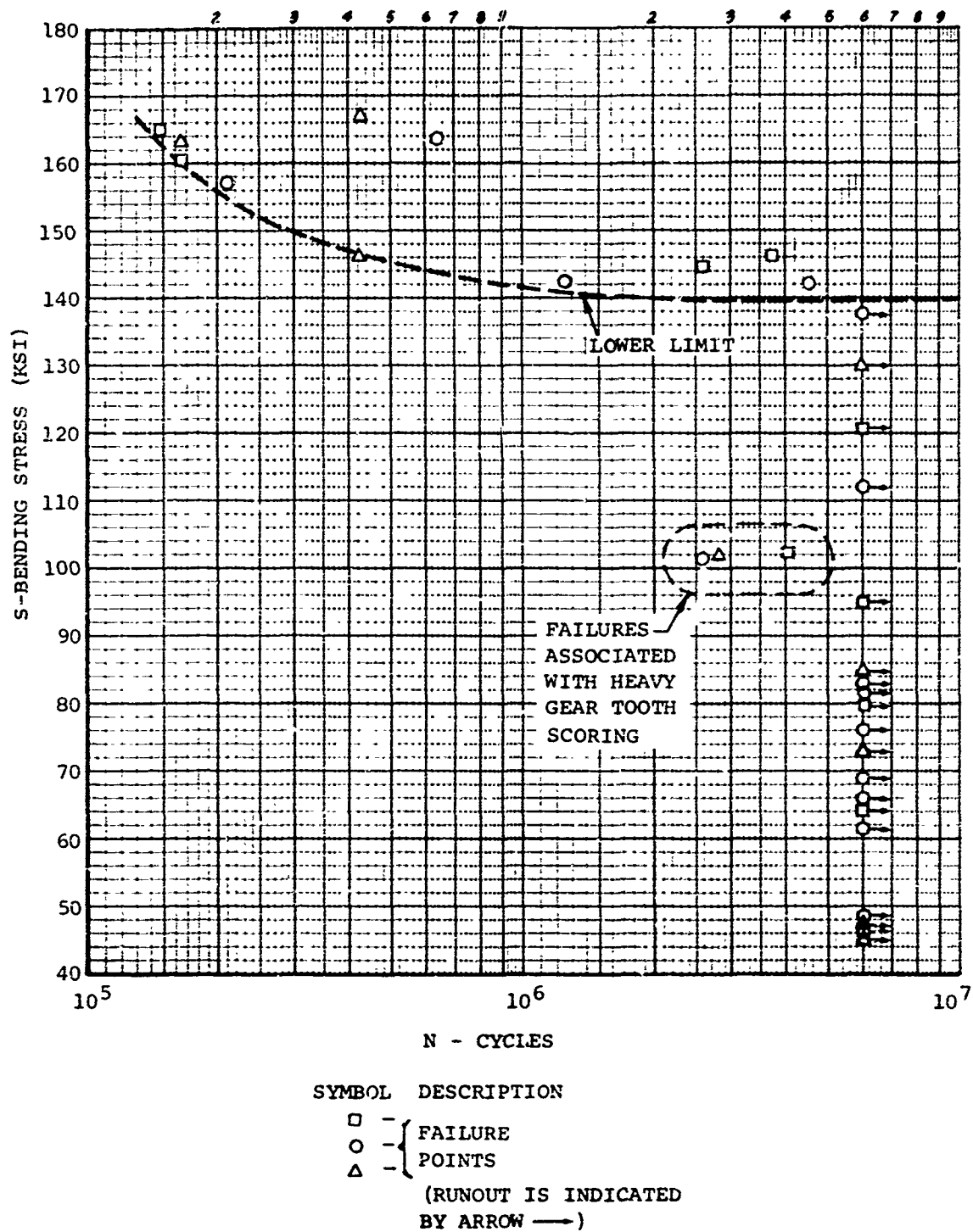


FIGURE 11. BENDING STRENGTH TEST RESULTS.

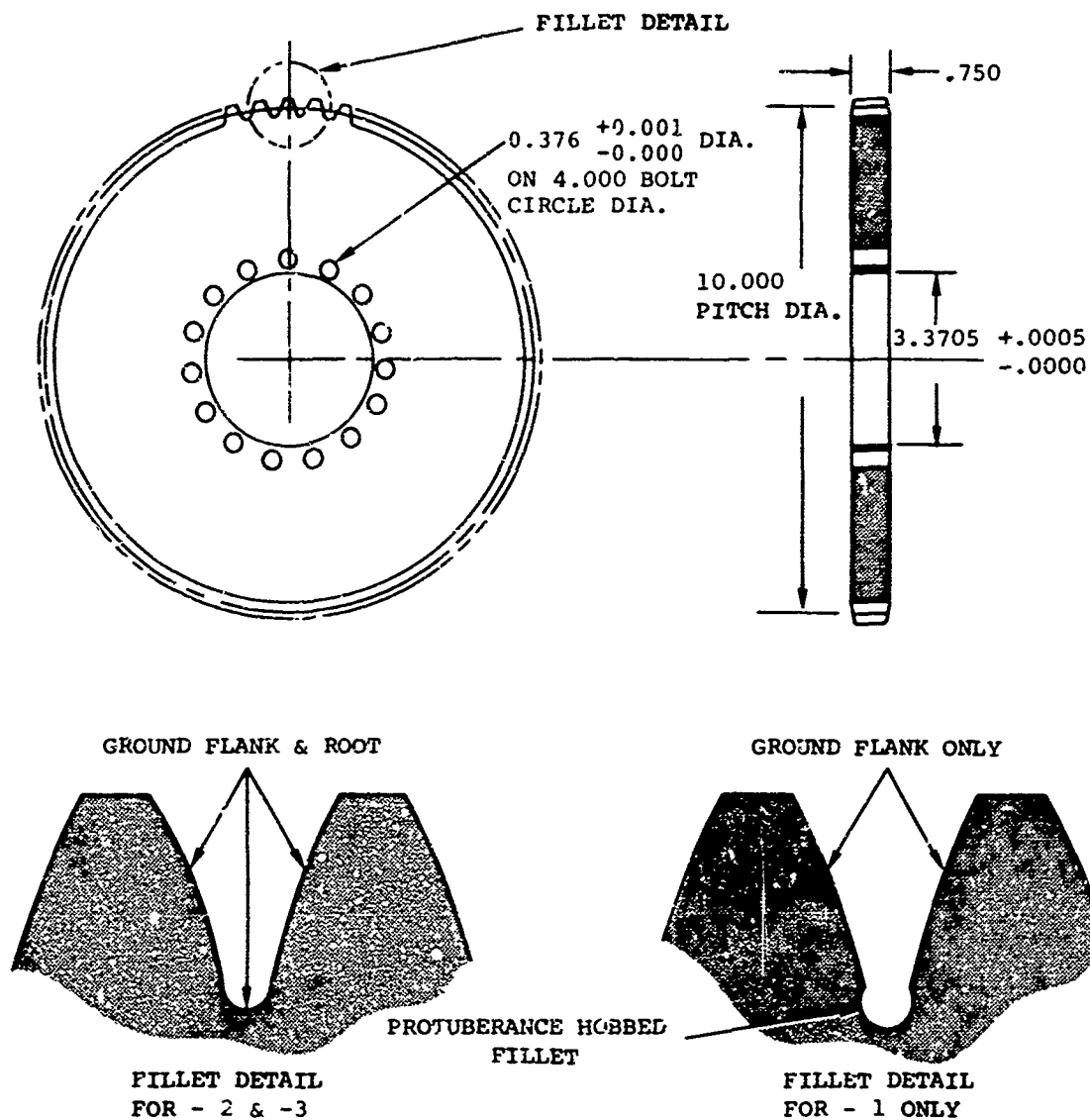
The gearing used was representative of the material and processing used in helicopter drive systems. (See Figure 12 for test specimen characteristics.) Because further improvement can be expected by development, it is believed that safe working bending stress can be realistically increased for the 1970 era.

In this program, failure data were obtained economically and expeditiously by using a gear research test stand. Most aircraft gear testing is not purposely continued to bending fatigue failure, especially in a gear case designed to accept the loading without distortion. This type of investigation is recommended for continuation, as are means of achieving improved strength-weight ratios.

Gear contact capacity is another consideration for gear sizing, and is of the same importance as bending strength. Both must advance by equal amounts to provide weight savings in gearing. It is believed that present design contact stress is nearer the permissible limit than is present design bending stress. This is evidenced by the frequency of contact distress phenomena occurrences as compared to the incidence of tooth failures. However, it is believed that several promising means of increasing contact capacity will lead to gains in this area commensurate with the gains to be expected in bending capacity. In consequence, the design of the HLH spur and helical gearing uses an index of bending capacity 50 percent in excess of that now currently used, and a similar index for contact capacity 20 percent in excess of that now used. These increases are mutually compatible in producing a balanced improvement in gear size.

### Bevel Gears

The allowable stress for a given gear type, as pointed out above, is an index number for that type, within a certain environment. It is not, therefore, necessary that the bevel gear stress number improvement duplicate that for parallel shaft gearing. After consultation with the Gleason Gear Works, and review of Vertol Division experience, it was decided to predict a 12 percent increase in bending strength capability, and a similar increase in surface wear capability as goals attainable with the aid of a bevel gear advancement program. Figure 13 illustrates a statistical analysis of



GEAR DATA: TYPE - SPUR  
 NUMBER TEETH -70  
 DIAMETRAL PITCH -7  
 PRESSURE ANGLE -25°  
 MAXIMUM LEAD ERROR .0003 IN/IN  
 MAXIMUM TOOTH TO TOOTH  
 SPACING ERROR -.0003

FIGURE 12. SPECIMEN - GEAR BENDING TEST.

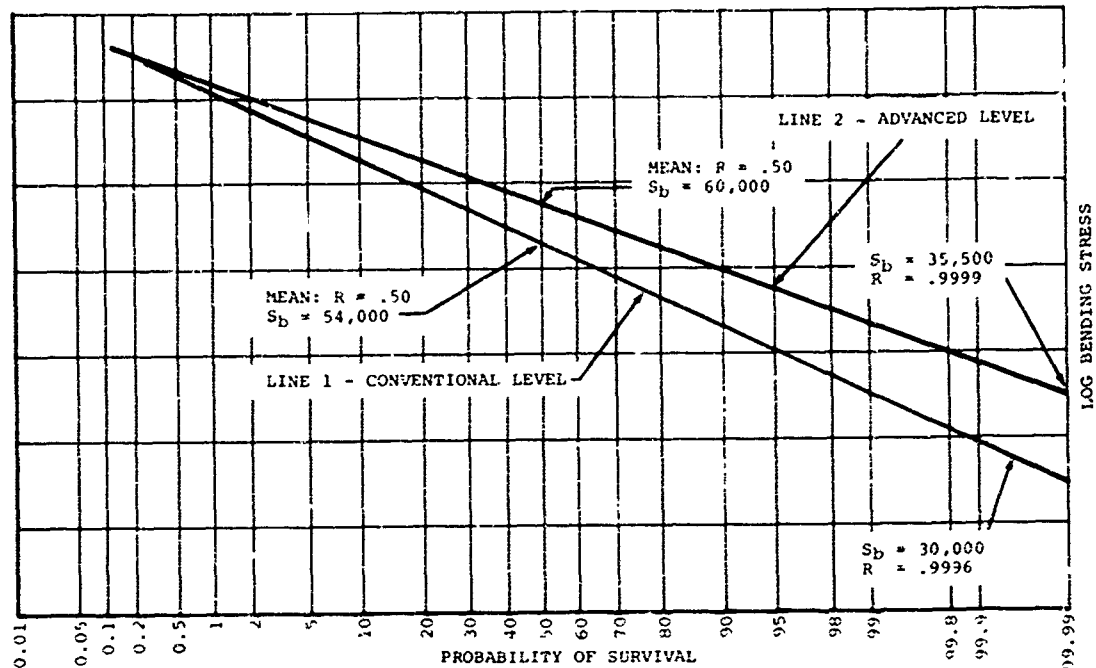


FIGURE 13. SPIRAL BEVEL GEAR BENDING STRESS VERSUS PROBABILITY OF SURVIVAL

TABLE III. GEAR STRESS LEVEL TABULATION

Item	Conventional	Advanced
<u>Bevel Gear*</u>		
Bending Allowable - (psi)	30,000	35,000
Contact Allowable - (psi)	225,000	250,000
Load Distribution Factor	1.1	1.1
<u>Spur Gear**</u>		
Bending Allowable - (psi)	30,000	45,000
Contact Allowable - (psi)	150,000	185,000

\*Calculation by Gleason tooth geometry and gear rating program.

\*\*Calculations:

$$\begin{aligned}
 \text{Bending } (S_b) &= \frac{W_t P_d}{F Y_k} \\
 \text{Contact } (S_c) &= 3180 \sqrt{\frac{W_t}{F \sin 2\phi} \frac{R + r}{Rr}} \\
 \text{Scoring } (T_f) &= \text{AGMA Flash Temperature Scoring Index} \\
 &217.01
 \end{aligned}$$

bevel gear bending fatigue stress related to reliability. The basic data is drawn from Reference 8. Line 1 includes all data points, and is the basis for the conventional stress level of 30,000 psi at  $R = 0.9996$ . Line 2 was drawn from the same data after discarding the lowest failure points which, by reason of material and processing, are not representative of HLH gearing technology. Approximately 10 percent of the failures were thus discarded to obtain a 35,500 psi stress level at  $R = 0.9999$ , which was the design assumption for the advanced stress level.

#### POWERPLANT SELECTION

The primary emphasis in this study was on the solution of the HLH transmission system, beginning at the engine shaft. The engine output connection was considered as the interface between engine and transmission.

Investigation of engines was therefore concentrated on the following:

1. A review of available engines, in the horsepower range applicable to two-, three-, or four-engine installations in the HLH, as presented by major manufacturers: General Electric, Lycoming, Allison, and Pratt and Whitney
2. Determination of numbers of engines required to perform the defined HLH mission
3. Preliminary design studies for each major engine considered, to determine the effect of its characteristics on the transmission arrangement and on the total reduction ratio requirement
4. Liaison with the engine manufacturers during the layout design stage, and discussion with them of major design problems

A tabulation of the available engines is given in Table IV. The design criteria for engine power was determined by the 95-degree 6000-foot altitude hover with payload, required for the transport mission (Figure 14). The effects of one-engine-out on the transport mission at standard conditions repre-

TABLE IV. POWERPLANT TABULATION

Mfg. Model Designat.		SHP (with SFC)			CHARACTERISTICS				COMPARISON FOR FLH						
		10 Minute		Military	Normal	Type	RPM	Drive Position	Weight	No. of Engines	Weight	Δ SHP for Critical Miss.(1)	Δ SHP for 1-eng-out Miss.(2)		
Lycoming LTC4B-11		--	2640	3400 (.533)	2300 (.566)	3030 (.544)	--	Free	16,000	Front	640	4	2560	-40	+400
X		--	--	3800	--	--	--	Free	--	Front	640	-	--		
General S4A Electric			2650	3400 (.488)	2450 (.500)	2935 (.499)		Free	13,600	Front	783	4	3130	0	+400
S5A				4500	2980			Free	13,600	Front		-	--		
GE-1				9500 (.464)	7385 (.472)	Cruise 4750 (.516)		Free	(3) 10,500 (4) 8,400	Front	(5) 1355	2	2710	+4170	-300
Allison T-78		--	--	(6)	(6)			Fixed		Front		3		(6)	(6)
548-C2		--	--	(6)	(6)		(6)	Free				3		(6)	(6)
T-56 M-25		--	--	6000	4200			Fixed	13,000	Front		3			
501 M26		--	--	5450 (.479)	3715 (.500)			Free	13,000			3	3100	+545	+550

NOTES: (1) Critical mission is transport hover at 6000 ft - 95°F

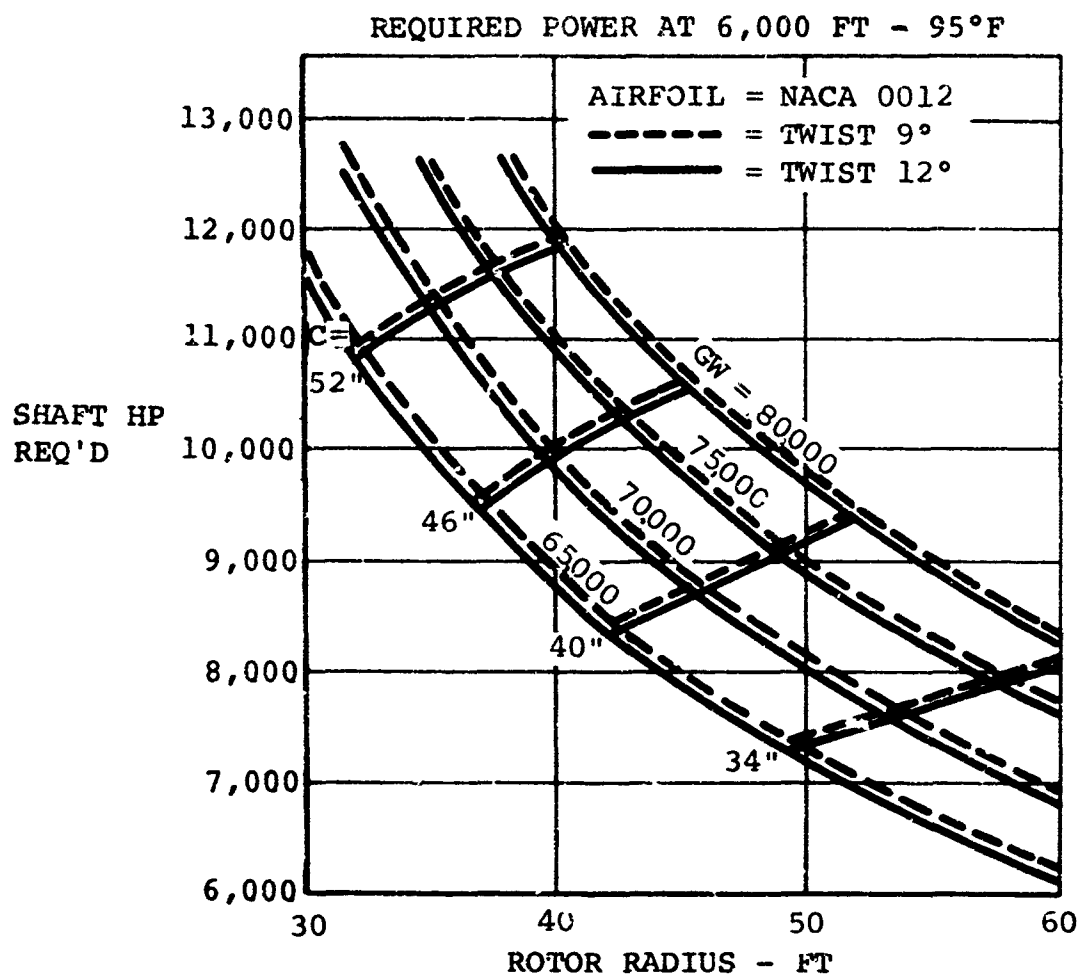
(2) Hover at SL Std condition with 12-ton payload and one engine out

(3) At MRP

(4) At cruise

(5) Includes oil cooler, no oil tank

(6) SHP and SFC are included in Confidential specification



BASIC ACCESSORY POWER	=	50 HP
DOWNLOAD FACTOR	=	1.04
XMSN EFFICIENCY	=	96%
NUMBER OF BLADES	=	3
OVERLAP	=	31%
TIP SPEED	=	670 FPS
CT/T	=	.1

FIGURE 14. POWER REQUIREMENTS.

sented less severe criteria, in the case of three- or four-engine installations. After examination of possible combinations, it was decided to limit the study to consideration of three- or four-engine installations, in the 5000-shp and 3800-shp classes, respectively. Two engines fall within each classification: the T-78 and T-56 in the 5000-shp class; the T-64 and T-55 in the 3800-shp class.

The performance of engines within a class was not sufficiently analyzed to permit a choice of one particular engine as optimum for the HLH. The delineation of a particular engine in the drive system design studies should not be construed as indicating preference for that engine, because, in fact, no such choice was attempted.

In the investigation of a three- versus four-engine configuration, however, some general characteristics were noted which influenced the final design toward four engines. HLH power requirements limited the three-engine design choice to the T-78 and T-56 engine families. These are presently configured as fixed-shaft, not free-turbine, engines. A rotor engagement clutch is therefore necessary for helicopter use. A review of engagement clutch design with Allison, and consideration of Vertol experience with the Allison-powered H-16A and H-16B helicopters indicated to Vertol that such a mechanism, while feasible, would represent a mandatory development item. It would also present the transmission system with certain requirements, notably, the provision of oil flow and oil cooling during the clutch engagement cycle. These requirements would impose penalties upon the transmission lubrication system.

A review of the fixed-shaft engine power-speed characteristics seemed to indicate a less favorable response to power demand than was available in free-turbine engines. The combined problems of clutch and power characteristics tended to place the fixed-shaft engine in an unfavorable position, relative to the free turbine.

The availability of free-turbine versions of these engines was discussed. It was understood that if these were offered, they would be in a rear-drive configuration only. This possibility was examined, and it was concluded that a three-engine, rear-drive requirement would add to transmission

complexity to the extent of requiring at least one additional bevel gear transmission. For this reason, the three-engine, T-56 family version, while able to meet the mission power requirements, was not continued into the detail study area. The T-78 family, while showing attractive weights and specifics, was not able to meet the mission power requirement; for this reason, it was dropped from the study.

The preliminary examination of transmission systems compatible with three engines did indicate that such a transmission system would meet problems similar in description and magnitude to the four-engine systems, with the added requirement of the engagement clutch.

The four-engine configuration was confined, as stated, to consideration of the T-64 and T-55. It will be noted that the planned development of both follows similar lines, and that the standard-conditions output horsepower predicted is 3800 in the 1970 period. The transmission system was designed to accept this power, although it is beyond the maximum requirements of the mission specification for the aircraft used as the study vehicle. The T-55B-11 and the T-64-S4A engines could provide 10-minute-rating power sufficient to maintain the HLH transport mission hover. (This hover requirement is for three minutes at 6000-foot, 95°F conditions.) The required power could be provided within the 3400-shp design. It is understood that both manufacturers could provide the 10-minute rating in advance of the 3800-shp military-rated design. Figure 15 indicates possible engine growth timing.

The major problem which must be explored before a four-engine (or three-engine) HLH is designed is in the engine control area. As explained in Recommendations, it is felt that this can be done by simulation, and that various control modes can be investigated sufficiently to provide confidence in the HLH configuration. Improvements in engine control on present two-engine production aircraft are currently under investigation by Vertol Division. The results of this investigation may provide assistance to the HLH program.

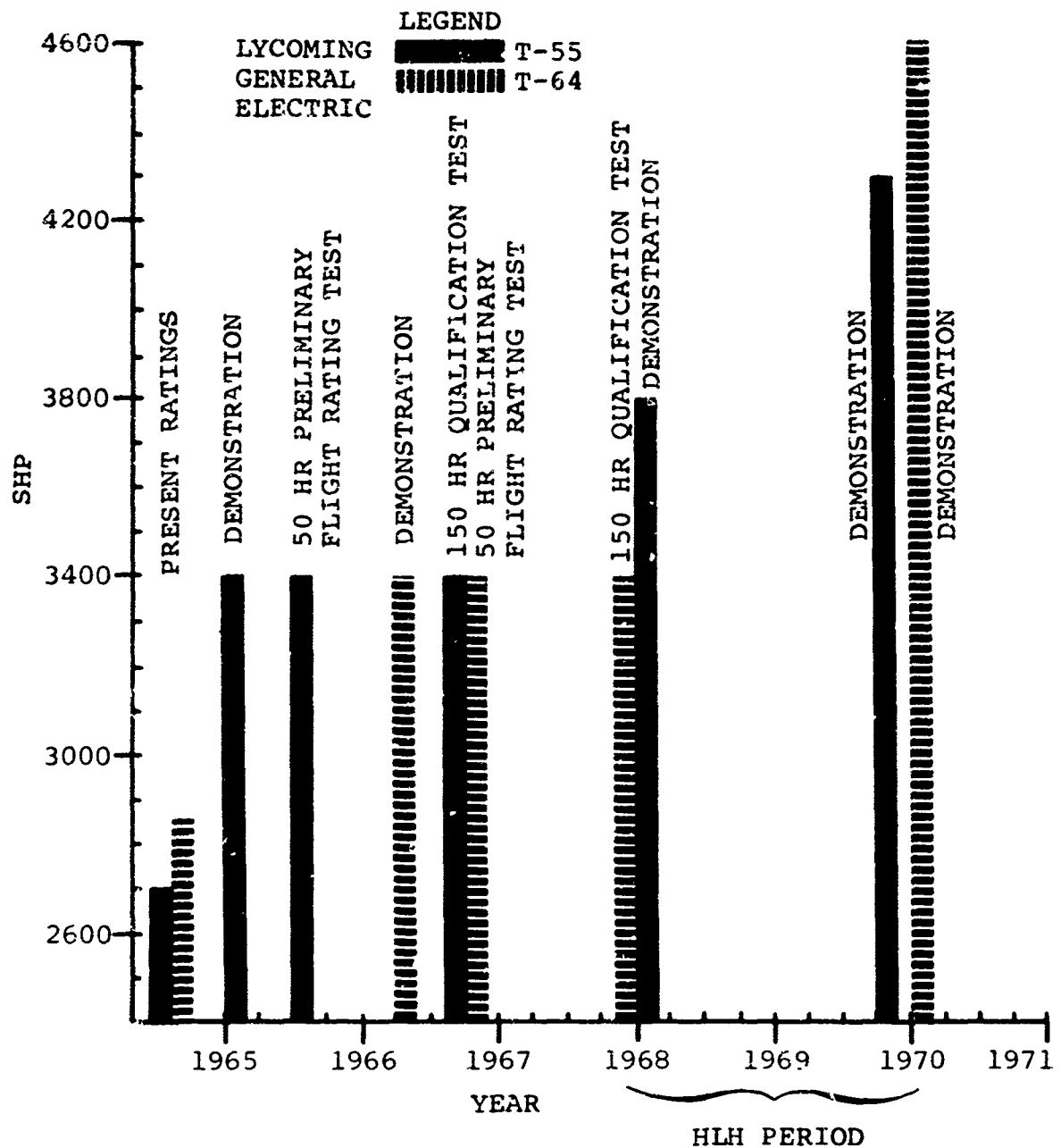


FIGURE 15. ENGINE AVAILABILITY AND GROWTH PLANNING (BASED ON ANTICIPATED FUNDING).

## TRANSMISSION SYSTEM CONFIGURATION ANALYSIS

In this section, the factors governing selection of a transmission system are outlined, the method of selection is described, and comments are made on six configurations selected for preliminary study. Vertol Division experience in the design and production of helicopters makes for full recognition of the interdependence existing between the airframe and the dynamic systems. This experience, as well as the experience gained in previous investigations and reviews of drive arrangements and components, has been used in the selection of the drive systems for preliminary study.

### Methodology

To systematize the possible approaches to be considered during this study, a comprehensive study of drive system arrangements was made. This included all the realistic approaches that could be visualized toward distributing power from two, three, or four engines to two rotors. These approaches were reviewed and grouped by such major drive system criteria as: number of gears, number of transmissions, and number of power meshes, to provide a ranking order. The number of power meshes reflects upon transmission weight as well as reliability. The power lost in a gear mesh is turned to heat, instead of lifting payload. At normal power loading, a single extra mesh in the HLH may be equivalent to nearly nine hundred pounds, and will contribute this amount to the total effective weight of the drive system. Total effective weight is derived as follows:

$$\begin{aligned} \text{Total effective weight} &= (\text{dead weight}) + \\ &\quad (\text{mesh loss equivalent weight}) \end{aligned} \quad (8)$$

$$\begin{aligned} \text{Mesh loss equivalent weight} &= \text{number of meshes} \times \\ &\quad \text{horsepower per mesh} \times \text{power loss} \times \text{power loading} \end{aligned} \quad (9)$$

$$\text{Example: } 1 \times 15,200 \times 0.0075 \times 7.5 = 850 \quad (10)$$

$$\text{Power loss} = \frac{\text{input power} - \text{output power}}{\text{input power}} \quad (11)$$

$$\text{Power loading} = \frac{\text{installed power}}{\text{aircraft gross weight}} \quad (12)$$

Other criteria used to indicate the final study configurations, but not considered amenable to a simple number value are:

1. State-of-the-art advances required in such areas as velocities, sizes, and loads
2. Effect of engine position on inlet ducting, exhaust position, reingestion, and accessibility
3. Comparative reliability, as indicated by overall complexity and by position of combining gears within the rotor synchronizing loop
4. Structural requirements necessary to mount engines and transmissions
5. Aerodynamic effect of necessary protrusions and housings
6. Maintenance of aircraft center-of-gravity within acceptable limits

Within the comprehensive group of configurations, it was evident that many mechanical problems were similar; typical component arrangements reasserted themselves as solutions to these problems. System studies were chosen to include as many of these arrangements as possible. This was done so that a number of solutions could be synthesized for problems encountered in the continuous process of refining the HLH aircraft and drive system.

#### Discussion of Configurations

The configurations discussed in this section are shown in Figure 34. In addition to the schemes discussed, this illustration also presents some alternate versions which were considered prior to selection of study configurations.

Major divisions of the schemes considered were established by the number of engines and by engine location.

Two-engine versions may be derived from arrangements which group two engines forward and two aft. Alternate arrangements to many of the versions include various positions of combining gears, an aft planetary located at the top of the pylon, and minor changes in gear ratios and intermediate speeds. These are not shown.

The configurations are ranked by mesh loss although, as explained, this is only one of a number of judgement criteria. Increased mesh loss is generally indicative, however, of increased weight and complexity and decreased reliability. Figure 16 indicates a partial list of the criteria used to select the study systems.

#### Four-Engine Configurations

Configuration I was chosen on the basis of low losses and desirable simplicity (Figure 2). This configuration and its variants is discussed in the RESULTS section, and illustrated by Figures 37 through 43. A version of this basic configuration (referred to as IA) divides the aft transmission into a lower bevel drive, and a final reduction atop the pylon. It is otherwise identical to the parent configuration.

Configuration IV was selected for limited study on the basis of being directly representative of a CH-47A transmission system, enlarged, and with two added engines. The advantages center about engine location; they are ideally positioned for direct inflow, ducts are not required, and the possibility of exhaust reingestion can be inferred as minimal. The podded installation would provide maximum accessibility. To confer these advantages, the transmission system increases in complexity. The number of gearboxes is increased by five; of this number, four may be identical engine nose boxes which, however, present particular problems in remote oil supply and scavenging. The rotor gearboxes are identical to those of Configuration I. This configuration has not been examined in detail; Figure 50 delineates the arrangement and intermediate speeds. Reference 7 includes a study of a system similar except for horsepower.

Configuration V was selected for limited study, particularly in the area of the combining transmission (Figures 51 and 52). This is a particular problem, in that, with a parallel engine arrangement, the center distances of the input pinions are established at such a distance that idler gears are required except at large reduction ratios (6 to 1 or more). Without idlers, gear pitch line velocity becomes untenably high at lower ratios. The insertion of idlers results in increased weight, mesh losses, and complexity. The combiner transmission shown demonstrates both approaches by showing one

design with idlers, and one without. In this application, the reduction ratio is such that it represents a borderline case for the use of idlers. With this configuration, a two-stage planetary can be used above the combining gears. The arrangement does not represent a simplification over Configuration I, nor does it reduce the number of bevel gears. Further objections are the difficulty of access to the engines, and to the vertical shaft, which runs through the center of the engine pack. With a two-engine input at each pylon, the problem of accessibility would be eased. The combiner transmission would be basically the same but with two inputs. The transmission system comments would pertain equally to this minor variation. The four-engine-aft configuration is depicted in Figure 51 with appropriate shaft speeds and gear ratios.

Configuration VI was selected for limited study, a major variation on the parallel-engine concept. In this case, the engines have been staggered to reduce the required combiner box input spans. A skewed engine arrangement was investigated at the same time, also with the objective of reducing combiner size, and eliminating idler gears. The skewed arrangement, in which engine inputs converged on a common point, was discarded when the necessary size of the engine compartment was established. Configuration VI, shown in Figure 53, represents a more attractive solution to the transmission system and to engine compartmentation. There is a major problem apparent, however, in the provision of satisfactory air intakes to the engines. The extreme aft position of the engines indicates that a reasonable aircraft CG location may be difficult to attain; in addition, penalties may be incurred in providing adequate mounting structure.

Configuration II was selected as a major study configuration. This arrangement is discussed in the RESULTS section. It is illustrated by Figures 44 through 47.

Configuration III was selected as a major study configuration in order to investigate the effect of a dual-shaft system on size and weight. It is shown in Figures 48 and 49, and is discussed in the RESULTS section.

Configuration X is another arrangement of dual shaft and rear

FACTOR	SYSTEM CONFIGURATION					
	I	II	III	IV	V	VI
TRANSMISSION COUNT	○	○	○	●	⊕	○
GEAR COUNT	○	⊕	●	●	○	⊕
POWER LOSS	⊕	○	⊕	●	⊕	●
COMPONENT VELOCITY (MAX.)	○	○	○	●	⊕	⊕
USE OF PROVEN CONCEPTS	○	⊕	⊕	⊕	●	●
TOTAL EFFECTIVE WEIGHT	⊕	○	⊕	●	⊕	●
ACCESSIBILITY OF DRIVE SYSTEM	○	○	○	○	●	⊕
ACCESSIBILITY OF POWER PLANTS	⊕	⊕	⊕	○	●	●
EXHAUST RECIRCULATION	○	●	⊕	○	●	⊕
ENGINE AIR INTAKES	⊕	⊕	●	○	⊕	●
AIRFRAME COMPLEXITY	⊕	●	⊕	⊕	●	●
ADDITIONS TO MINIMUM OUTBOARD PROFILE	●	●	⊕	⊕	○	○

COMPARATIVE RATINGS

○ SUPERIOR

⊕ AVERAGE

● INFERIOR

FIGURE 16. COMPARISON OF DRIVE SYSTEM CONFIGURATIONS.

mounted engines and represents a dualized version of Configuration I. It illustrates the penalties of direct translation from single shaft to dual shaft, in that the mesh loss and number of gears are increased. This is avoided in Configuration III by arranging two engines with a direct drive into the interconnect shaft and combining the other two into the rotor transmission.

### Three-Engine Configurations

Three-engine transmission systems were given limited study to determine whether specific problems, and specific advantages, accrue to the use of three engines. As noted in POWERPLANT SELECTION, the major study effort was concentrated on a four-engine vehicle.

Configuration XII has three engines arranged in a "T" formation, combining upon one gear. The low power loss, and number of bevel gears (6) make this an attractive schematic arrangement. Limited study of this system revealed that velocity of components directly related to the input section increases by approximately 10 percent as compared to the four-engined Configuration I. These components include bevel gears, overrunning clutch, and bearings. Such an increase does not necessarily represent a crossover into problem areas for these components, although, as noted previously, the overrunning clutch requires development effort.

Aside from the input section, the remainder of the three-engine transmission systems investigated show no unique problems. The engagement clutch has been reviewed under POWER-PLANT SELECTION and represents a major development item, if the use of fixed-shaft engines is contemplated. The rotor transmission final reductions shown in Configuration XII, for example, are identical in arrangement and size to those used in Study Configuration II.

### FINAL REDUCTION SUBSYSTEM ANALYSIS

In this section, various methods of obtaining the final reduction of speed to rotor rpm are considered. The factors leading to a choice of method are outlined, and the different approaches are compared.

### Subsystem Requirements

The final reduction is required to deliver power to the rotor shaft efficiently, reliably, and within a compact outline. The proportion of the total system ratio reduction that it accepts is dependent upon matching its own characteristics with those of the drive system ahead of it. Because no method of obtaining the entire reduction in one package (without multiplication of gear stages) is now available to the helicopter designer, the practice is generally to take ratio reductions wherever a direction-changing gear mesh must exist in the system. The final reduction is left with the residual ratio which for various reasons was not extracted in the preceding meshes. This residuum is typically 15 to 30 percent of the total reduction, in the medium transport helicopter.

The HLH configurations under study have total system ratio requirements of 100 : 1 to 150 : 1, depending upon engine selection. This is roughly twice the CH-47A ratio. The gearing upstream of the final reduction was arranged to give either one or two meshes in the different study configurations. By using the ratio reduction potential of these meshes to the fullest extent, the requirement for final reduction ratio was set at approximately 45 to 1 (with one preceding mesh), and 20 to 1 (with two preceding meshes). This selection took into account present state of the art in final reduction methods. Other methods now under development will, if successful, influence a choice to a higher final ratio in order to keep intermediate speeds higher, and torque and weight lower.

To conform to the state-of-the-art requirement, all power transmission was by involute-tooth-form gearing, although requirements exist to which the non-involute tooth form (conformal) could be applied with advantage.

It is more economical, from a subsystem weight standpoint, to reduce any given ratio by load-sharing meshes, than by transmitting all load through one mesh. As an example of the weight characteristics of single-mesh versus parallel-mesh subsystems, reference is made to Figure 17. This indicates the weight saving effect of using a number of parallel meshes to replace one. It is evident that while three or four parallel meshes decrease weight significantly, a

diminishing improvement is obtained from more than this number. Using assumptions other than shown, the optimum number will change, but the change will not be substantial. In addition to the accepted benefits of multiple meshing, which mostly relate to the desirability of packing in the most working metal within a given envelope, these systems lend themselves to epicyclic (planetary) motion, with the added advantage of increase in ratio, and decrease in power loss. Therefore, the final reduction investigations concentrated on parallel mesh and epicyclic motions.

The ratio requirement of under 20 to 1 was well suited to the proven ability of the two-stage epicyclic approach. The assembly for Configuration I, shown in Figure 39, draws heavily upon the developed CH-47A planetary system. Gear mounting, ratio division between first and second stages, and component sizing are the result of this successful experience. This solution, therefore, has a high confidence level. It is worthy of note that the HLH under study does not require a new concept of final reduction in order to obtain an effective drive system.

Systems which combine engines on a gear which is immediately adjacent to the final reduction input (Configuration II, and others) require a higher numerical ratio than the foregoing. To achieve the required 45 to 1 ratio with parallel-mesh, epicyclic methods, an investigation of arrangements was conducted. The results are given in the following four studies:

1. Compound first stage and single second stage (Figure 54)
2. Three stages (Figure 55)
3. Split-torque first stage and single second stage (Figure 56)
4. Roller gear, three stages (Figure 57)

Schematic representations of the four types are shown in Figure 18.

The results of the study are shown in Figure 19. Based upon total effective weight, the chosen method, which is entirely within the state of the art, is the compound and single configuration SK 14232. This is shown in Configuration II. The roller gear represents an attractive alternative, if

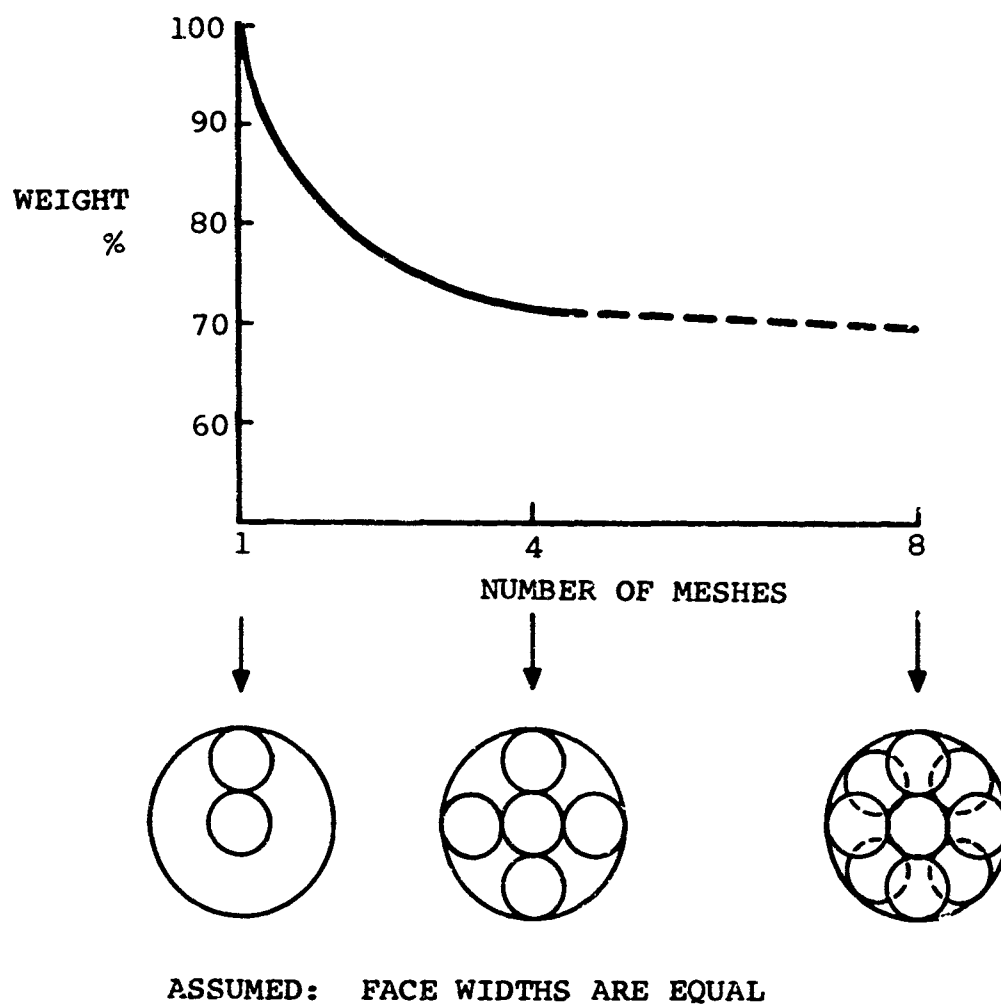
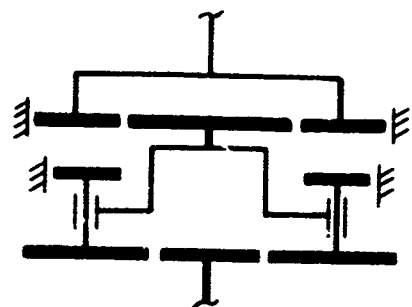
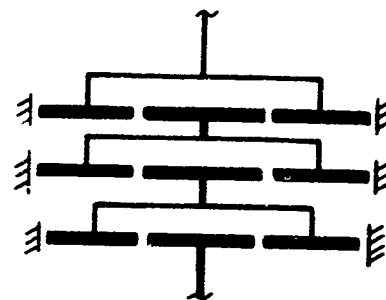


FIGURE 17. EFFECT OF PAPPALLEL MESHES ON WEIGHT.



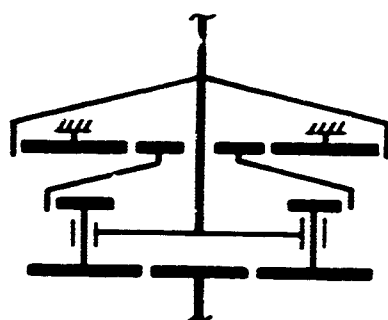
COMPOUND AND  
SINGLE STAGE  
SK 14232

①



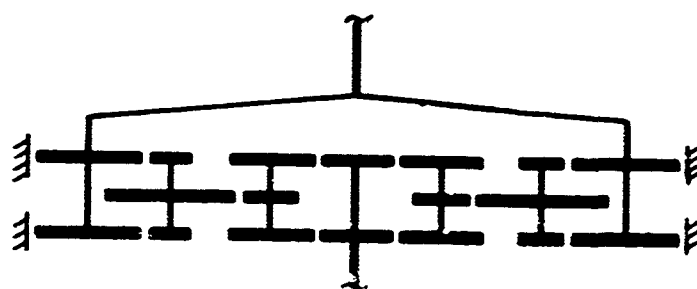
3-STAGE PLANETARY  
SK 14231

②



SPLIT TORQUE  
SK 14230

③



ROLLER DRIVE SK 14233  
(EXTENDED FOR CLARITY)

④

FIGURE 18. FINAL REDUCTION METHODS - SCHEMATIC COMPARISON.

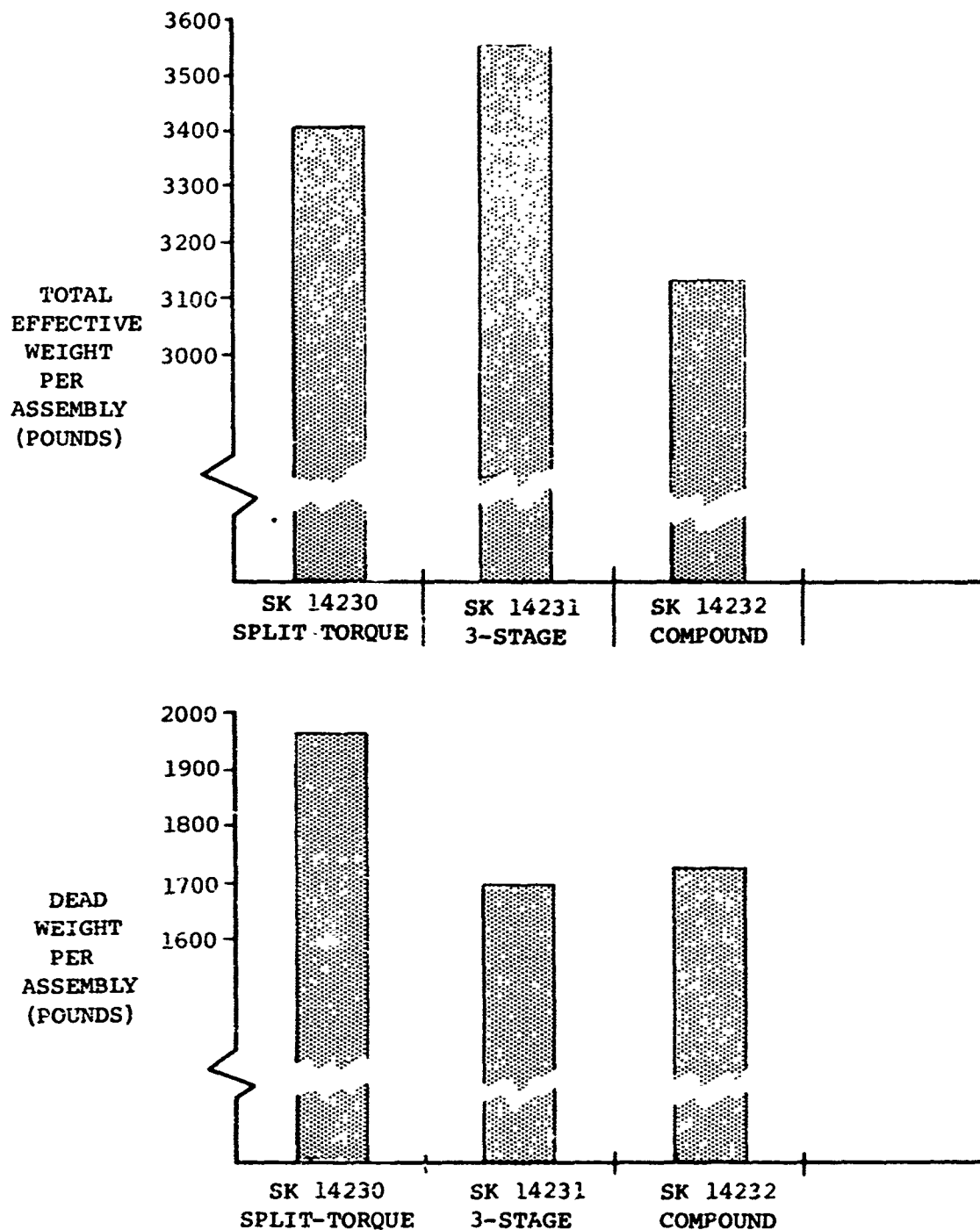


FIGURE 19. WEIGHT COMPARISON OF REDUCTION METHODS.

further testing continues to uphold early model-test indications. Comments on each arrangement are grouped below under the main criteria of choice:

#### Power Loss

The "equivalent number of meshes" were calculated according to the method of Buckingham (Reference 1) and others, to obtain the power losses. Because an epicyclic system entails relative motion of gear centers, the velocity of engagement is the sum of tangential and gear-center (orbital) velocities. Mesh loss is the product of load and velocity; therefore, a lowered resultant gear mesh velocity gives a smaller power loss than a fixed (nonorbiting) system. This is generally true of the epicyclic systems used in helicopter drives. On the other hand, it is possible in a differential system to increase velocity, so that the resultant power loss is far above that which would be obtained if the gears were fixed.

The effect of relative velocity of the gear elements was taken into account in the analysis of the methods. An apparently small change in mesh loss can significantly affect the total effective weight of the method, and therefore, its competitive position. For example, the three-stage planetary, with a low dead weight but a comparatively high power loss, is relegated to a low position in overall weight standing.

To reduce the dead weight of the roller drive, a further reduction stage was added. This is shown in Figure 18 as Arrangement 4. This arrangement was investigated as an epicyclic, with fixed ring gear. The mesh loss is slightly less than that of a fixed-center system, because of the orbital motion in the first and final stages. However, the orbital speed, equivalent to output rpm, of the gears contributes little to reduction of first-stage resultant velocity and mesh loss. In the same way, the split-torque method does not gain significantly in mesh loss because the carrier is rotating at output rpm (at this ratio, 2 percent of input rpm). This is practically a fixed condition when compared to a simple planetary, where orbital rpm may be

30 percent of input rpm, with the effective mesh loss substantially reduced in consequence.

It is significant that the lowest power losses for a 45 to 1 method are not greatly in excess of those for the 20 to 1 reduction. Therefore, the higher ratio is practical from a total effective weight standpoint. It is, however, close to the practical upper limit for the methods investigated, unless an added stage is used. Figure 20 shows equivalent meshes and range of ratio for the methods studied. Refer also to equations in Appendix I.

#### Volume and Weight

The maximum diameter of the final reductions was determined by the ring gear required for the last planet stage. The load carried by this stage is equivalent to rotor torque. In the arrangements shown (Figure 18), it is distributed over six planet gears.

The split-torque design (Arrangement 4) necessitates a rotary ring gear. Enclosing this gear will necessitate a gear case from four to six inches larger than would be required for designs using a fixed ring. To react load, the split-torque design shown uses the planet carrier. There is an inherent weight penalty in that the split-torque arrangement requires all the components of the fixed-ring designs plus a large disc, to make the transition from ring gear to rotor shaft. Similarly, in the first stage the rotating ring gear requires a substantial disc to carry out load.

Arrangements 1 and 3 use essentially similar first stages. Clearance diameter for the rotating cluster gears is less than the final ring gear because of the ratio split chosen. However, because only three planet gears can be geometrically accommodated, the first stage is heavy and deep, by comparison with the six-gear final stage.

To reduce first-stage bulk, it was separated into two stages; this resulted in Arrangement 2. A lower weight is possible because more planet gears can be fitted in,

to share load, when the ratio is lower. Arrangement 2, the three-stage planetary is lowest in volume and in dead weight.

In order to fit in more parallel-mesh gears at a given ratio, the roller gear drive uses offset planet gears. The primary weight advantages inherent in this approach are: first, that the second and any succeeding stages benefit by two meshes per gear; and second, that the second-stage pinion is driven from both ends, thus reducing torsional deflection in this small-diameter, wide-face gear. Because of the small diameter, it is possible to obtain ratios of 35 to 1 (as shown in Figure 57) with three meshes, in a reasonably compact design.

A roller gear ratio of 45 to 1, consistent with the other designs, was investigated. It is desirable to limit the diameter of the final reduction unit. To accomplish this in two stages, pinions of small diameter and correspondingly large-face width were required. A further effort was then made to compact the 45:1 assembly by adding another stage (Figure 18). The diameter of the ring gear was reduced, as would be expected, at the cost of additional mesh loss. Because of the relatively small amount of operational experience with this drive, and because time did not permit optimization of the design, no final weight estimate was made. If results of planned full-scale tests are favorable, the device should be at least competitive in weight with more conventional systems. To demonstrate superiority in total effective weight, it would be necessary to establish, from full-scale testing, that the method of positioning results in higher efficiency or greater gear stress allowables than are obtained with conventional planetaries at the same ratio.

#### Load Sharing

To gain from parallel meshes, all paths must share in the load. If all paths could be induced to carry exactly equal loads, under all conditions, a significant improvement in load-carrying ability of the assembly would be realized. The magnitude of inequality is approximated from calculation and experimental evidence.

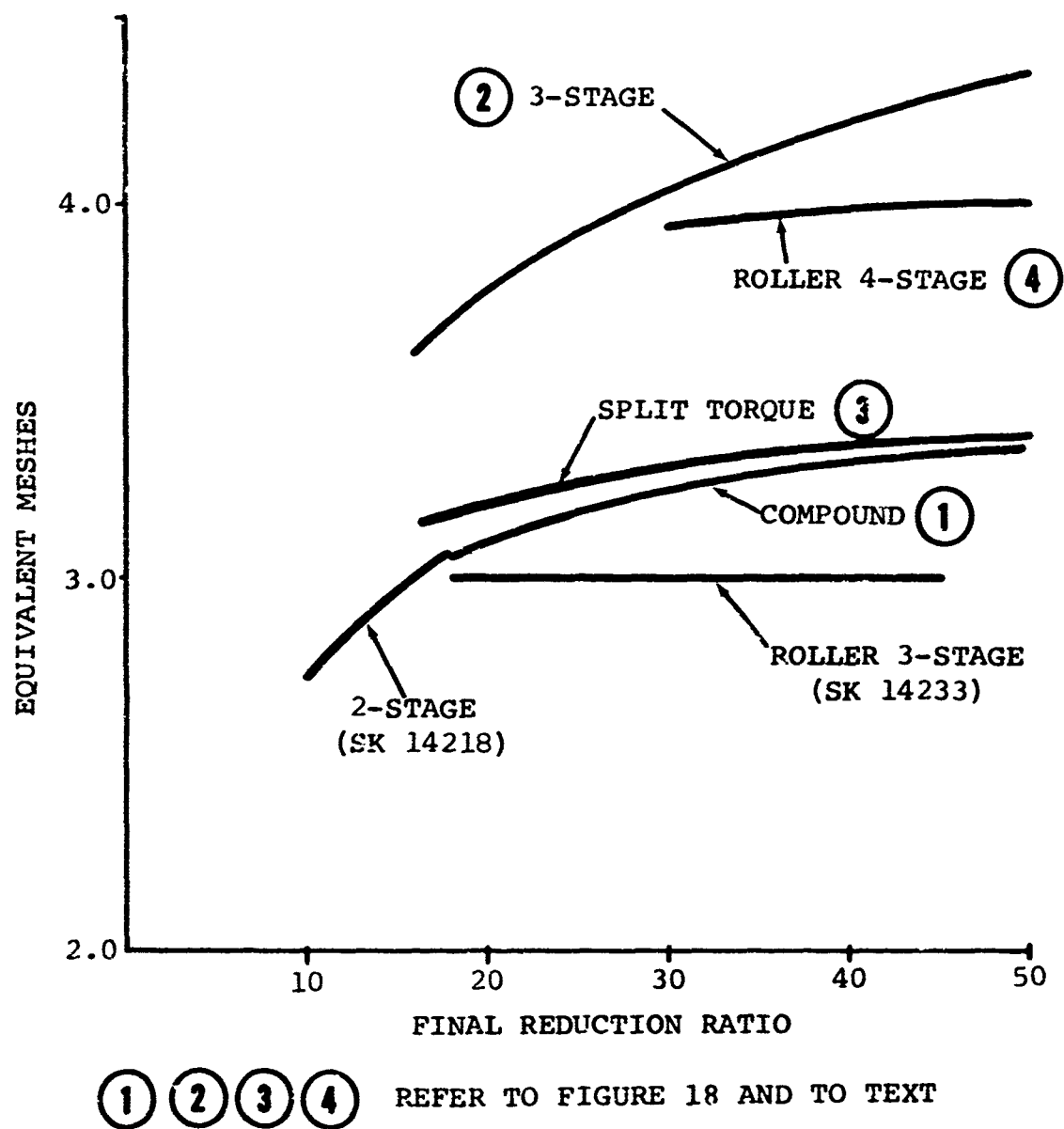


FIGURE 20. EQUIVALENT MESHES OF FINAL REDUCTION METHODS.

Load sharing is highly dependent upon component accuracy. It may be improved in some cases by the following approaches:

1. Controlled elasticity of the supporting members, as for example, the planet carrier posts
2. Mechanical freedom in members; for example, floating (spline-supported) sun gears, and/or ring gears which are allowed to shift radially under load
3. Equalization of axial forces among a group of (helical) gears, which will automatically equalize tangential forces. (This is particularly applicable to a compound planetary such as SK 14232 (Figure 54); various methods, including use of hydraulic devices, have been proposed.)
4. Mechanical equalizing levers or cams to allow planet gears to adjust position to inequalities in tangential load

Approaches 1, 2 and 3 are either in use, or appear especially suitable for helicopter final drives. Approach 1 has been shown in the drawings.

To share load in a different sense, that is, between stages, the split-torque method has been suggested. Analysis of the split-torque design shown in Figure 56 (SK 14230) disclosed that although 25 percent of the torque was in fact transmitted from the first-stage carrier direct to the output shaft, the benefit to the second stage was minimal. This was due to the non-orbiting design of the second stage, whereby its reduction ratio was numerically less than a geometrically similar orbiting arrangement. This meant that the input rpm to this stage was necessarily less, to accomplish the same overall ratio. Therefore, the torque, for the same power, was greater. This effectively negated the torque split in the first stage, and resulted in second-stage gear sizes which are the same as those used in the other designs.

The primary advantage seen in the split-torque device is that it reduces orbital velocity of the first-stage

planets. This is an important consideration when the input rpm approaches power turbine speed. The centrifugal effect is not, however, significant in the speed range shown here, where centrifugal force represents only 10 percent of the load upon the first-stage planet bearing.

#### Conclusion of Design Comments

The designs shown in Figures 54, 55, 56, and 57 uniformly reflect the choice of stress levels proposed for study for the HLH time period. If such designs were to be constructed before implementing a complete evaluation program, a more conservative approach would necessarily be taken. The changes in size and weight would be of the same order as that shown by comparison of Figures 9 and 10. The single most important size effect would be growth of the final stage planetary diameter by 15 percent. This would place the ring gear in a size range above that now accommodated in heat treatment presses and gear grinding machinery. The ring gear designed to advanced stress levels could be handled with existing production equipment. Based on present experience, the heat treatment distortion expected in a carburized ring gear of either diameter will create manufacturing problems. To reduce these, an investigation of materials and processes should be conducted. One suggested approach is the use of through-hardening steels, to eliminate the carburizing cycle.

#### BEVEL GEARING CONSIDERATIONS

The HLH drive system, by reason of the horsepower transmitted, requires that more-than-normal consideration be given to certain aspects of the bevel gearing design. These are: (1) mounting accuracy, (2) vibratory loads, and (3) scoring hazard.

#### Mounting Accuracy

Bevel gear design calculations for stress and load carrying capacity involve certain assumptions regarding the tooth bearing which will be obtained in the gear mountings under load. Unless these assumptions are closely fulfilled, actual stresses may be considerably different from the calculated values, and early failures may result. If the gear mountings

could be made very rigid, the behavior of the tooth bearing under load would be predictable and a suitable tooth bearing could be developed based on experience. In helicopter drive systems, however, mounting designs differ greatly, and rigidity is sacrificed to some extent in favor of weight.

The physical dimensions of the HLH transmission cases, and the magnitude of the gear loads imposed upon them, indicate that adequate mounting rigidity of the gears is more difficult to obtain than in smaller, less highly loaded transmissions. The larger gear teeth used in larger transmissions are not, unfortunately, tolerant of proportionately higher deflections. Therefore, emphasis must be placed on design approaches which are inherently rigid, and which therefore minimize weight.

The design approaches used throughout the HLH study have included:

1. Large-diameter, rigid, gear shafts
2. Straddle-mounting of the pinion member
3. Preloaded thrust bearings to reduce axial motion
4. Double-web support for large-diameter gears

#### Vibratory Loads

A gearing design may produce natural resonant frequencies in the gear and its supporting web and shaft which are within the spectrum of the vibratory inputs. A minimum-weight design, operating at higher meshing frequencies, is particularly susceptible to this problem, which, if not checked, is likely to cause catastrophic failure. The primary causation for a given gear set is forcing frequency; tooth stress level, rim velocity, and gear diameter do not, taken singly, cause a resonance problem.

In the past, the technique for obtaining a solution to bevel gear rim and web resonance had not been well defined. Previous investigations of parallel shaft gearing (Reference 4) led, however, to a general understanding of the experimental techniques which could be used to indicate the natural frequencies of the gear on its support. With these known, a reliable appraisal of the existence and magnitude of the problem can be made by comparing the natural frequencies to

the range of operating frequencies. If the natural frequencies fall within this range, corrective action is required. One corrective action consists of attempting to stiffen the gear support and raise the natural frequency by adding metal. It has been shown that this is not particularly effective, at least within the restrictions of a flight-weight transmission. A more powerful approach is to absorb the resonant energy by damping. This has been successfully accomplished in a number of instances. It represents a reliable and minimum-weight solution to the problem.

As previously noted, the level of stress in gearing is not, of itself, likely to predispose the gear set towards a resonance problem. It is not foreseen that the advanced stress levels proposed will make the problem more likely to occur, or more difficult to solve. In fact, by reducing gear tooth size, they may reduce the magnitude of the forcing function. However, it may be difficult to stiffen large-diameter gears sufficiently to avoid the entire range of forcing frequencies, notwithstanding the inherently rigid designs employed. Therefore, the HLH gearing will be tested for dynamic response in the design stage. This can be done readily by using semifinished or "boiler-plate" models. If damping is desirable, this too can be evaluated by the same relatively quick and inexpensive method. There is, therefore, a proven approach to the elimination of rim and web resonance problems; this approach will be applied to the HLH gearing.

#### Scoring Hazard

Scoring is defined here as the welding, and subsequent tearing out, of gear tooth asperities. It is a function of gear tooth load, lubricant film strength, velocity, and other factors which may raise local surface temperatures above the melting point. A severe scoring situation is rapidly evidenced, and is self-perpetuating. It is one of the boundary conditions which determine gear load-carrying capacity, and as such, is of the highest importance to the HLH drive system.

The load-carrying requirements of the HLH drive system necessitate relatively large tooth sizes, for bending strength. The size of the tooth is indicative of the sliding velocity, which is a factor in heat generation. The combination of coarse diametral pitch (large tooth sizes) and high unit

tooth load is conducive to the development of scoring. However, Vertol's past experience and that of others indicate that the scoring hazard can be overcome through the development of suitable surface treatments or deposited coatings such as silver plate, tuftriding, alphasizing, etc. Additional possibilities for resisting the scoring hazard exist through the development of improved surface finish and topography and the proportioning of tooth geometry to reduce the instantaneous temperature rise.

Designs using advanced tooth bending stresses will result in smaller tooth sizes. This will be conducive to lessening of scoring tendencies.

Indications, using the Gleason scoring index, are that the HLH bevel gearing, without modification, is within the scoring problem area. As noted, there are solutions which have been applied to other gearing, and which have been found effective. It is now necessary to establish the approach for the HLH requirements of load level and tooth geometry and size.

## RESULTS

In the previous section, TRANSMISSION SYSTEM CONFIGURATION ANALYSIS gave the background for the configurations that would be further studied. These were selected from among a large number of possible arrangements. This section is devoted to analysis and discussion of those systems chosen for further study.

By the requirements for this study, two configurations were to be pursued up to the point where a selection could be made; one configuration was to be completed.

As the study developed, it became apparent that several widely different arrangements should be considered to indicate direction for both transmission and airframe development. The minimum requirements of the study were therefore exceeded, in that two configurations were completed, and a third configuration was continued into a study of one gearbox.

### CONFIGURATION I

The transmission system which has been chosen as the one in which most confidence can be placed at this time, is Study Configuration I. This configuration was chosen because:

1. Engine arrangement in aft fuselage minimizes the problems expected from recirculation and proper inflow.
2. There is a minimum number of gearboxes (two).
3. Selection and placement of speed reduction mechanism results in final drive and bevel gearing arrangements which are extensions of CH-47A experience.

### Inherent Reliability

The combining of multiple engines by means of a single gear case forming part of the aft rotor transmission represents a substantial reduction in number of transmissions, complexity of lubrication system, and number of meshes, when compared to the configuration shown on Figure 50. The

arrangement also removes the combining function from the interconnect shaft, with an expected improvement in system reliability. Further study may prove the practicability of a breakaway section, or clutch, between combiner and rotor cases, to free the rotor system from the combining gears in case of a catastrophic failure of the latter. This would permit a safe autorotational descent with rotors synchronized.

Further, the rotor brake has been located on a shaft extension of the combining box, away from safety-of-flight shafting. The plane of the disc is clear of the accessories mounted above and forward, on top of the combining box.

#### Aerodynamic Consideration

The position of the engines, required to accomplish the transmission system simplifications enumerated, requires a stub wing extension from the base of the pylon. The structural and aerodynamic effects must receive more detailed analysis than is possible in a transmission study. However, to answer the most immediate question, preliminary analysis was made of hover download, intake efficiency of engine ducts, and exhaust recirculation probabilities. This investigation provided a satisfactory level of confidence in those areas.

#### Hover Download

The mean span of the housing (stub wing extension) for the four aft-mounted engines is about 28 percent of blade radius, based on a 48-foot total radius. With the present trend of 20-percent root cutout and slipstream contraction, hover download is significant only for the parts of the fuselage which are outside 25-percent radius from the rotor center. The download on the present arrangement is about 150 pounds for the two engine housings (Figure 21).

For the forward and aft arrangements, the engine housings are both less than 25-percent rotor radius from the centerline of rotation and do not, therefore, contribute significantly to the hover download.

For lower rotor radii, corresponding to higher disc loadings, the engine arrangements shown in Figures 2

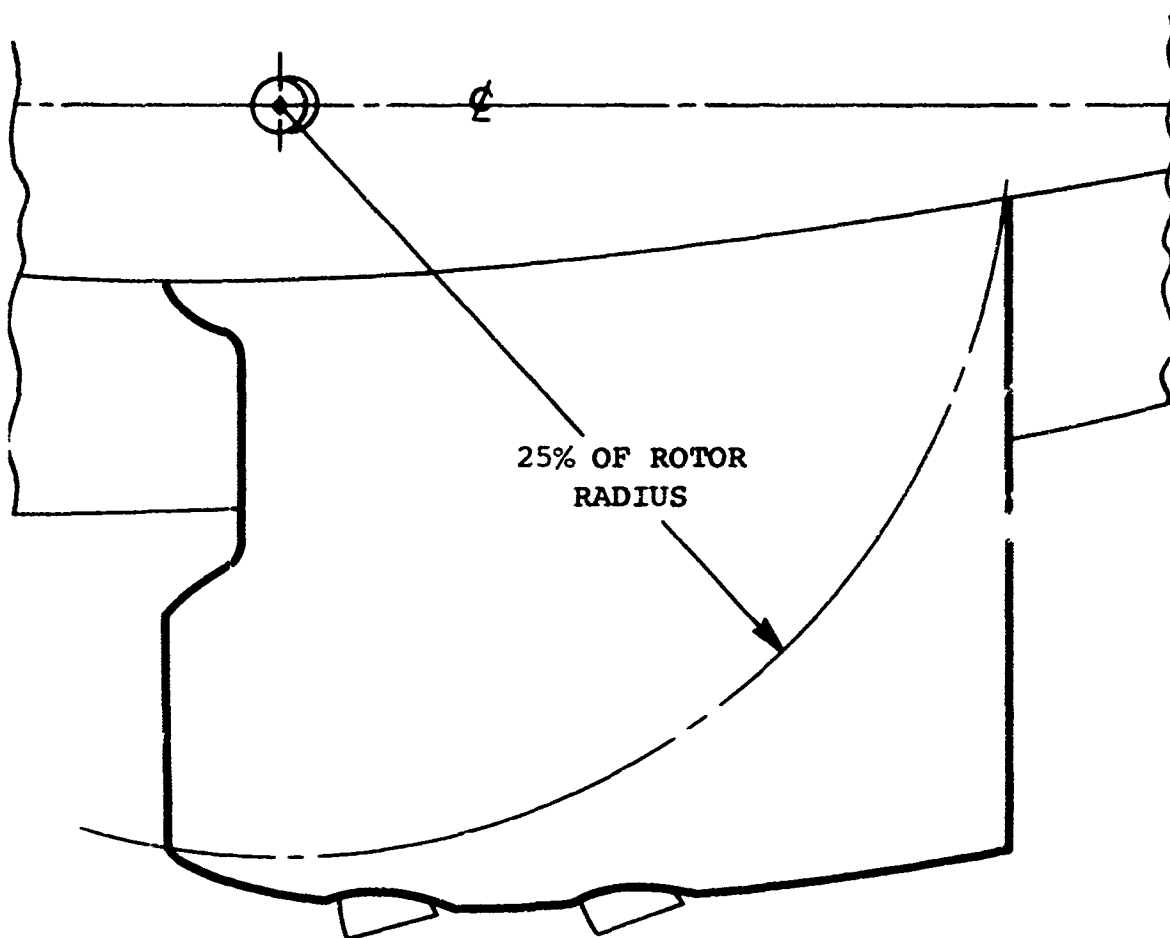


FIGURE 21. ENGINE HOUSING PLANFORM.

and 3 may be moved inboard. The air inlets for this configuration would be deepened to meet the mass flow requirements.

#### Inlet Ducting

Inlet losses are held to a minimum by the large intake ducts. Velocities in the ducts are of the order of 100 feet per second, thus presenting a low dynamic pressure to obstacles such as the drive shafts and to 90-degree turns in the ducting. At the engine inlets, the air velocity increases to approximately 250 feet per second, eliminating adverse pressure gradients due to diffusion.

In forward flight, the airflow external to the inlet ducting traverses an NACA Series I cowling lip on the leading edge of the airfoil-shaped wing. This lip is designed to prevent airflow separation on the wing during all flight conditions. Steep descents will provide the greatest angle of attack to the cowling lip. The lip geometry can be optimized by wind tunnel tests.

#### Exhaust Recirculation

In a configuration study, exhaust recirculation could be examined by the test technique described in Reference 6. The approach has been successfully used to determine optimum engine location to prevent possible impingement of exhaust gases on the aft pylon. Figure 22 illustrates one step in the investigation using smoke flow in a 1/8-scale helicopter model. An extension of this approach is believed applicable to determination of flow patterns from engine exhausts over various flight regimes. Engine exhaust recirculation would necessarily be predicted and corrected at an early stage in the final design.

#### Maintainability and Size of Components

To illustrate the accessibility features of Configuration I, Figures 23, 24, and 25 are included. These show the use of the engine housing as a work platform for engines and aft transmission. The engines are removable by lowering straight down from the installed position, as indicated on Figure 21.

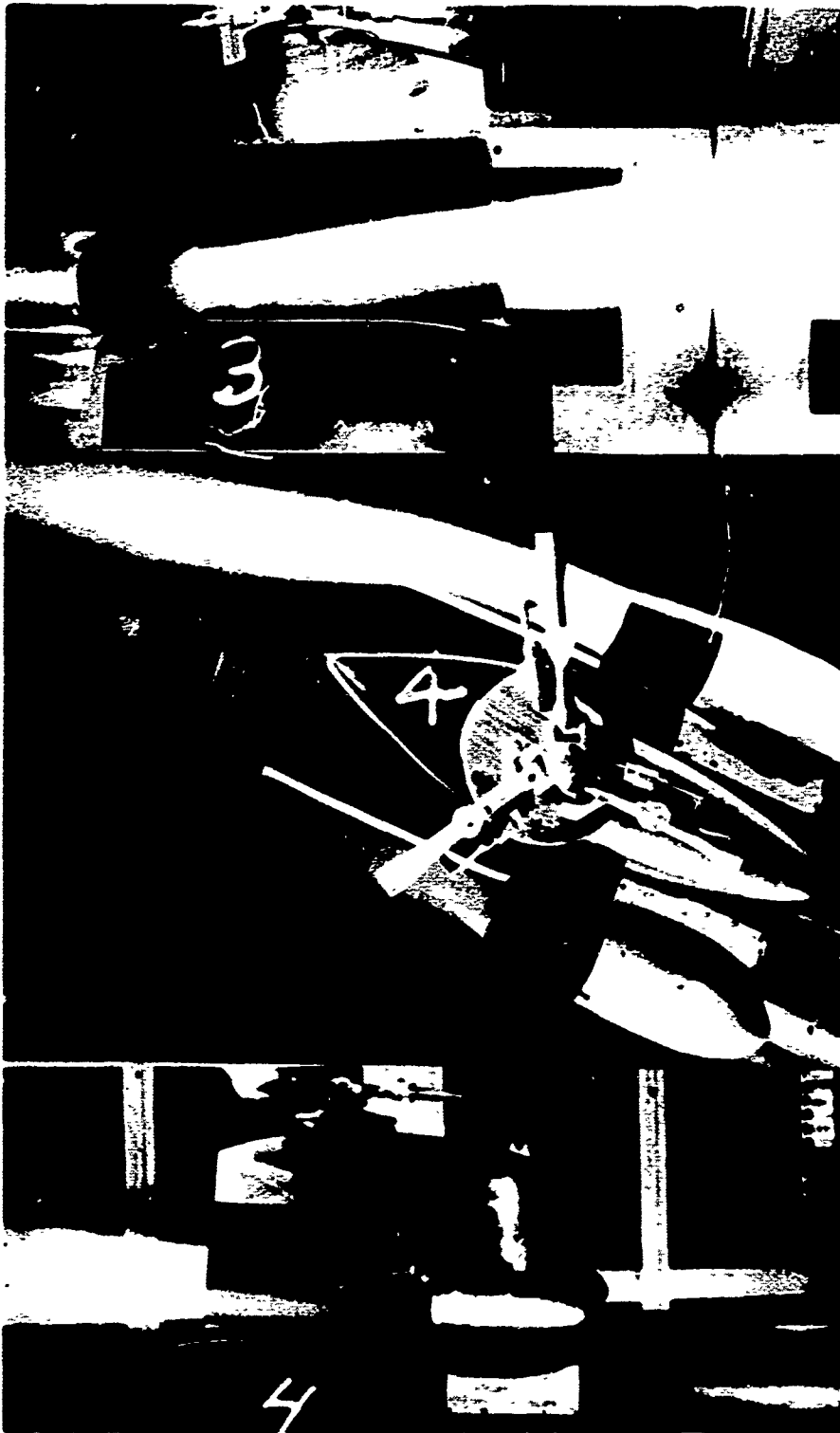


FIGURE 22. EXHAUST STUDIES.

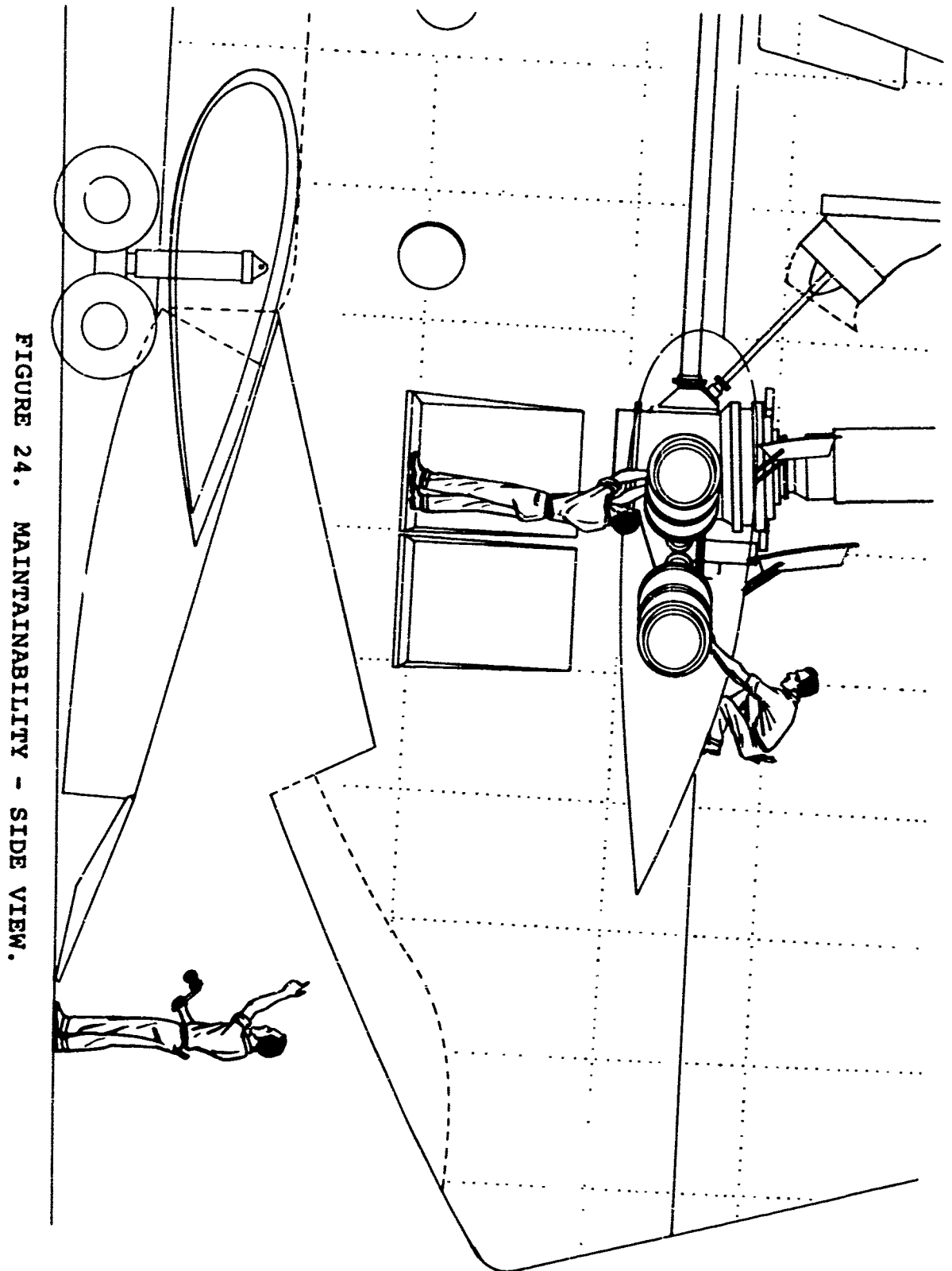


FIGURE 24. MAINTAINABILITY - SIDE VIEW.

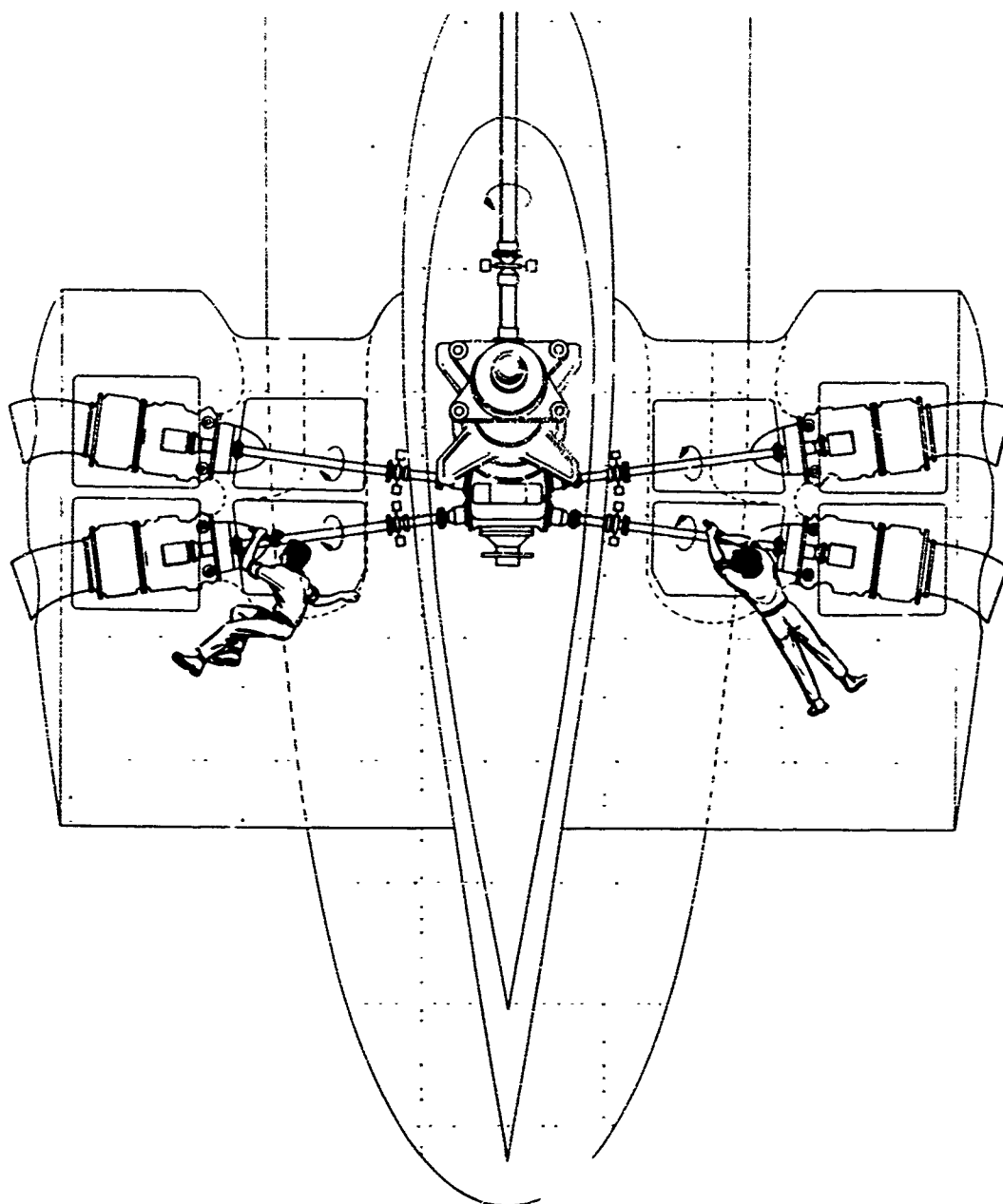


FIGURE 23. MAINTAINABILITY - PLAN VIEW.

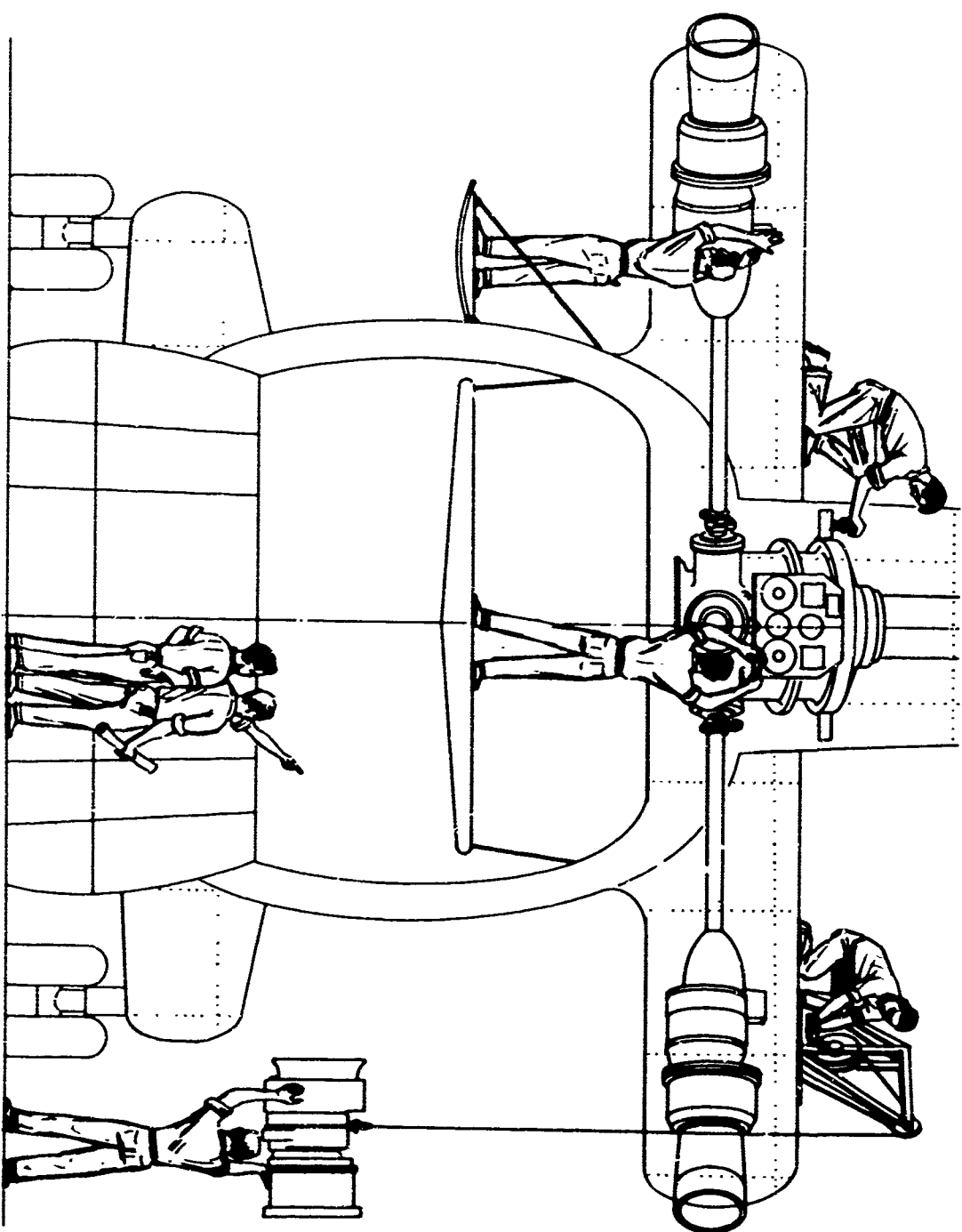


FIGURE 25. MAINTAINABILITY - VIEW LOOKING FORWARD.

Maintainability, although not covered in this report, is considered an important factor in the comparison and selection of a transmission design.

### Variations of Configuration I

A number of alternates have been investigated within the general framework of Configuration I. These were:

#### Supercritical Versus Subcritical Speed Shafting

The comparison is shown in Figures 26 and 27. At the same shaft rpm, and using similar diameters, the supercritical shaft shows advantage by weight reduction (160 pounds) and by reduction in number of sections (2 versus 10). This is despite the fact that a steadyrest support is placed in the center of the supercritical shaft to restrain shaft deflection under flight maneuver conditions. The work currently in progress at the Vertol Division under USAAVLABS sponsorship will, at its conclusion, provide additional information and an increased confidence level in the use of supercritical shaft designs in this application.

#### High-Speed Interconnect Shaft

An interconnect shaft system operating at engine output rpm was investigated for possible advantage over a medium-speed (6000 rpm) system. In both cases, the supercritical speed approach was used. The medium speed was selected for this configuration because of the following:

1. There was no change in total effective weight (Figure 27). A 9-percent increase in power loss, combined with a heavier final reduction, served to cancel the weight advantage of higher-speed, lower-torque shaft and bevel gears.
2. The rotor transmission becomes more complex and larger in volume to provide the additional ratio reduction necessary to the high-speed system.
3. The bevel gear pitch line velocity was substantially increased by the higher shaft speed.

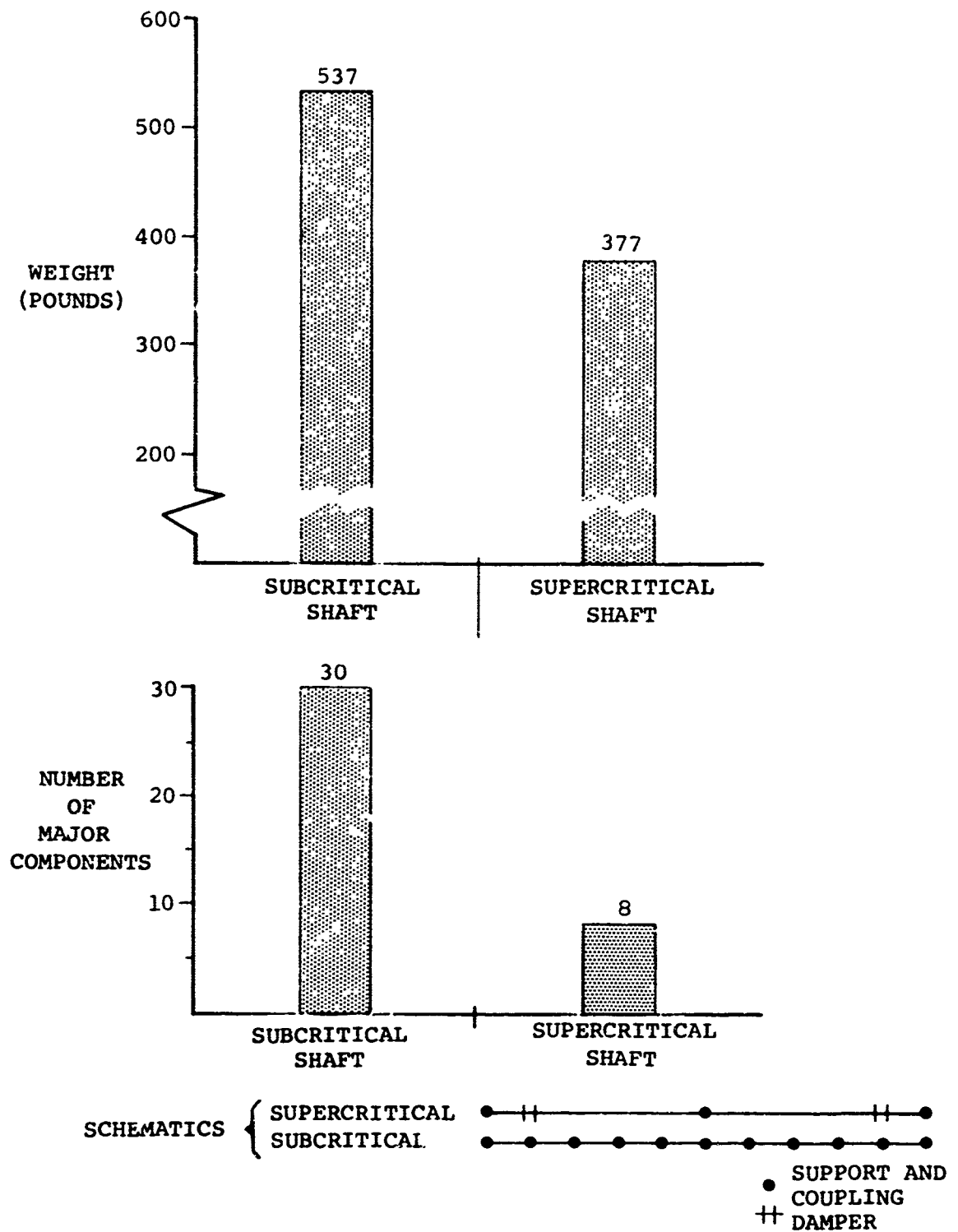


FIGURE 26. COMPARISON OF SUPERCRITICAL AND SUBCRITICAL SHAFTING.

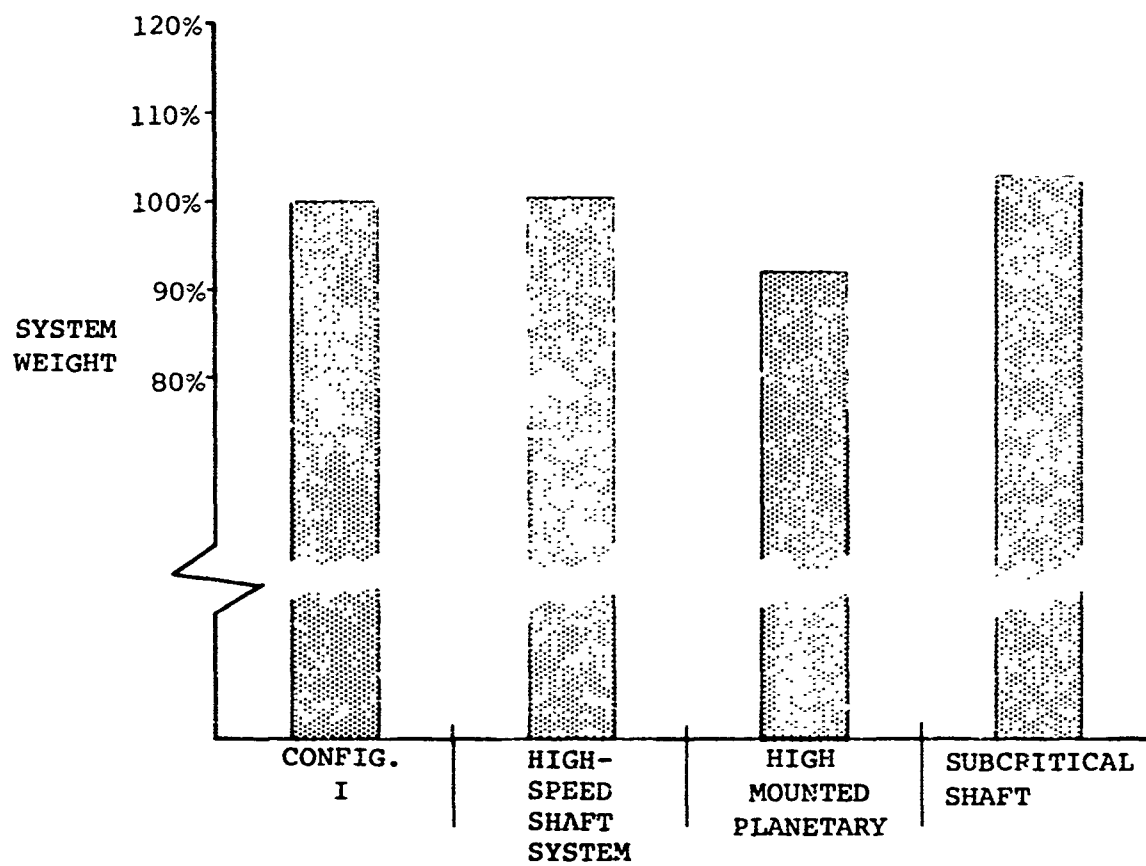


FIGURE 27. CONFIGURATION I - WEIGHT EFFECT OF ALTERNATE APPROACHES.

Maximum velocity is 180 percent of that selected for Configuration I.

#### Final Reduction Position

The final speed reduction gearing for the aft rotor may be mounted with the bevel gears at the pylon base (Figure 41), or separated from the bevel gears at the pylon top (Figures 42 and 43). The first approach provides the minimum number of gear cases (two). It reacts rotor torque at the pylon base, with a direct load path to the fuselage longitudinal structure. However, it requires a shaft above the final reduction gearing capable of transmitting rotor torque at rotor speed, and also capable of resisting the bending imposed by rotor loading. The alternative approach, to place the final reduction gearing at the top of the pylon, reduces the torque requirement and eliminates the bending from the rotor shaft. There is, in consequence, a substantial reduction in shaft size and weight. The reduction in weight chargeable to the transmission system is estimated at 7 percent of the system by this approach. To set against this, an additional transmission is, in effect, created by the separation of bevel gearing and planetary gearing. Torque reaction must be carried down the aft pylon with a possible increase in structural weight. The structural weight increase has not been included in the above 7 percent. It will be recalled that this alternative approach was used in the Vertol H-16 helicopters. Further study of the comparative advantages is indicated, and will be conducted before a final configuration is selected.

#### Effects of Engine Choice

The major effects upon the transmission system of a choice between the two engines, T-64 or T-55, are as follows:

1. A ratio change would be required because of the different engine output speeds. The T-64 would require an overall ratio of  $13,600 \div 134 = 101.5$  to 1. The T-55 would require an overall ratio of  $16,000 \div 134 = 119.5$  to 1. The

difference could be taken in the first reduction ratio (Configuration I), giving a 14-percent larger pinion for the T-64 than the T-55, because of higher input torque and lower ratio. Gear pitch line velocity would be approximately the same. The remaining system sizes and speeds would be unchanged.

2. The engine torquemeter, now a part of the T-64 output shaft casing, would be revised to agree with a new and longer output shaft casing.
3. To preserve the same direction of rotor rotation, the combining section of the aft transmission would be changed to place combiner gears on the opposite sides of their pinions. This would accommodate the opposite shaft rotations of the T-64 and T-55.

#### CONFIGURATION II

Any study of tandem-rotor helicopter engine installations must inevitably consider the merits of mounting engines on both the forward and aft pylons. Such an arrangement has been used in the past by Vertol Division on the H-16 series of helicopters. The installation of engines close-coupled to the rotors offers certain apparent and actual advantages; however, recent Vertol Division practice (as instanced by the 107 family) has been to group the engines at the rear pylon. The reasons for this are:

1. Problems of exhaust reingestion are minimized. The multitude of flight conditions under which the helicopter operates makes even the most straight-forward inlet-exhaust relationship a matter of concern and extensive study. Positioning of exhausts and intakes along the fuselage increases the complexity of the problem, and decreases the probability of a solution satisfactory for all flight regimes.
2. Forward location of the engines may increase noise, heat, and vibration levels in the cabin area. The penalty is increased shielding or damping material to counteract these.

3. Engine controls, fuel distribution, and servicing requirements are separated as far as is possible within the confines of the fuselage, thereby increasing weight and complexity. From a vulnerability viewpoint, this penalty might be admissible.

There are some possible advantages from forward and aft engines. The HLH concept on which this study is based has a particular requirement for a fuselage length greater than that of the 107 family, but comparable to that of the H-16. It would appear that the length of fuselage might increase the significance of such considerations as:

1. A normally unloaded interconnect shaft, with (under the worst conditions) a requirement for one-half the shaft torque capacity normally required of the all-aft-engine configuration.
2. Dispersion or dilution of the exhaust stream in its passage from forward to aft pylons.

In addition, at least one of the forward and aft engine configurations initially analyzed demonstrated that simplification could be achieved by this arrangement; it also showed the least number of gear meshes, and consequently the best effective weight, of all transmission system arrangements considered. Because of these benefits, it was decided to evaluate and compare this configuration (Configuration II) with Configuration I. This comparison could indicate whether the drive system advantage was sufficient to outweigh the expected additional complexity required in other systems by this arrangement. Comments on the system are as follows:

#### Component Arrangement

Engines are combined upon a bevel gear, which is (schematically) mounted upon the final drive input. This is a most simple and satisfactory arrangement which does, however, require a high-ratio, efficient, and compact final drive. The analysis of various forms of final drive is to be found in the FINAL REDUCTION SUBSYSTEM ANALYSIS section. Briefly, a compound planetary gearing arrangement was developed for this application. It is considered that no exorbitant weight or efficiency penalties were incurred, as compared to the two-

stage final reduction shown in Configuration I.

The interconnect shaft system, as was noted, operates unloaded when power supply and demand are equalized between rotors. The design case for this shaft takes into account an engines-out condition, so that shaft and bevels are designed with sufficient fatigue strength to accept the power of one engine. The important consideration is that, compared to Configuration I, the advantages of this unloaded shaft system are not substantial for the following reasons:

1. Supercritical speed shafting, used in both configurations, has reduced the weight of both shaft systems. For this reason, the difference in design loading between the two systems results in a reduced delta weight for Configuration II of approximately 1 percent of the transmission system weight.
2. The indeterminate load direction and unloaded cycles which the shafting, couplings, bevel gears, and splines would experience would not necessarily result in higher reliability for an unloaded system.

#### Inherent Reliability

This is, as indicated, not necessarily improved over Configuration I. It may be less, because of the greater complexity of the rotor transmission. The rotor transmissions incorporate the combining function; in Configuration I, this is delegated to a separate combining gear case which is not within the synchronizing loop.

#### CONFIGURATION III

It is necessary to continually increase helicopter drive system reliability. There are at least two general approaches to the achievement of this; they are:

1. Arrangement of the drive system to provide maximum simplicity, and reduction in the number of locations where the failure of one element can contribute to the loss of an aircraft
2. Detail design and processing of the transmission

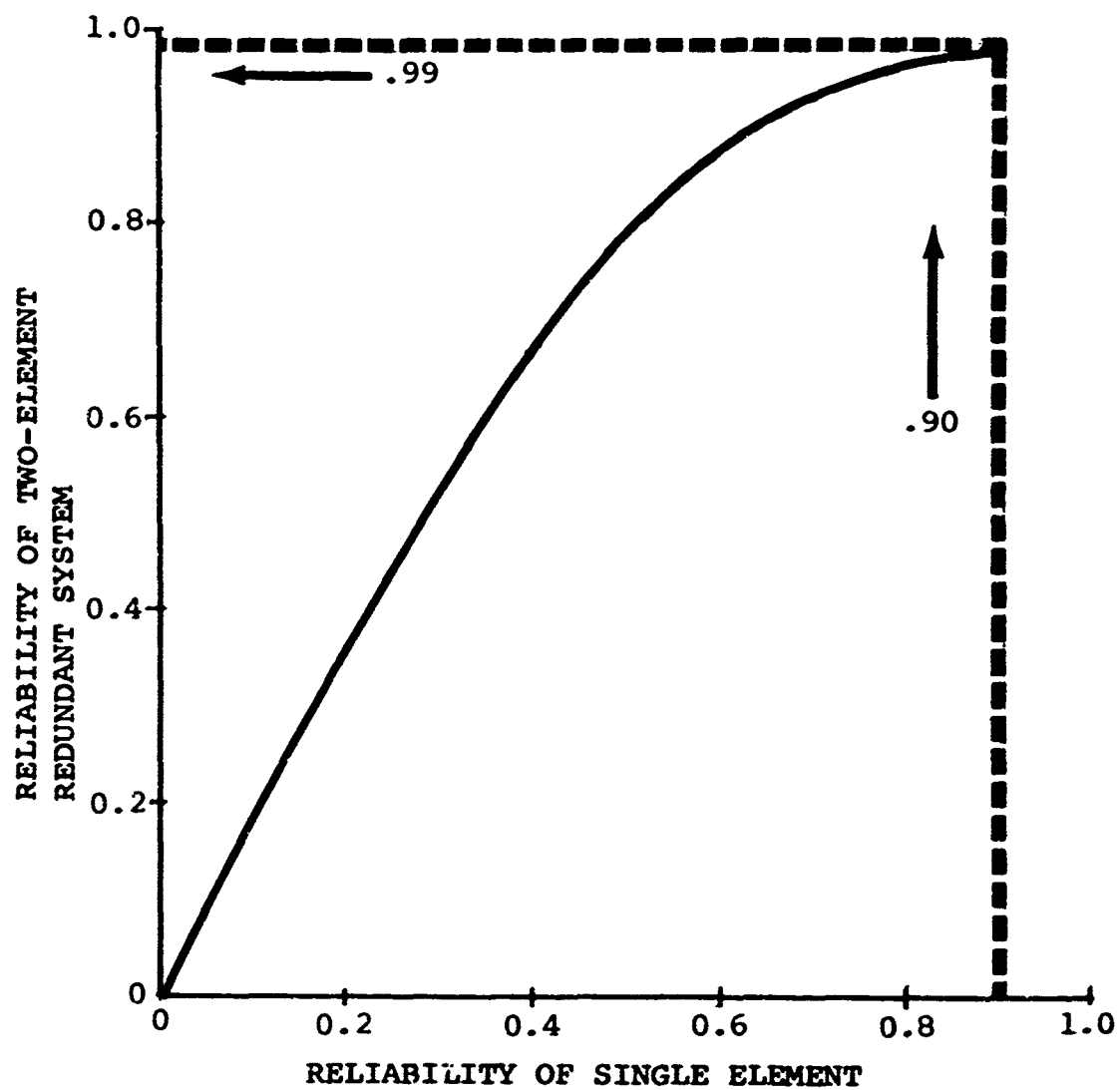
components to ensure predictable and uniform response to the environmental conditions to which they are subjected (Consideration must be given to the occurrence of environmental conditions which lie outside the expected range.)

It is to be expected that approach 1 will be met insofar as the separate, and sometimes conflicting, demands of transmission and airframe allow. Approach 2 is increasingly attainable as material technology and design experience advance. However, a condition may be foreseen wherein increasingly sophisticated manufacturing controls will produce decreasing improvements in reliability.

A third approach is therefore indicated. This is the use of parallel load paths, which yield theoretical improvements in reliability of the magnitude shown in Figure 28. The extent to which the actual improvement matches the mathematical prediction depends largely upon the independence of each load-carrying path. The load paths must be so designed and constrained that failure of one does not cause the failure of the other. If this is not done successfully, then the overall system reliability is degraded. The actual reliability improvement depends also upon inspectability of each load path.

Many instances already exist of multiple load paths purposefully applied to modern helicopter drive systems, and more applications are continually under study and development. While recognizing this and taking advantage of it in all study designs, it was thought proper to extend the approach and to investigate an HLH drive system which dualizes the interconnect shafting and spiral bevel gearing. This is delineated in Study Configuration III. The objective is to maintain complete interconnection of the rotors, despite the failure of a shaft, coupling, or bevel gear. The following comments can be made on this configuration:

1. The 32-inch separation of the two interconnect shafts allows inspection of each, and permits structure between them to act as a barrier in case of failure of one.
2. Spiral bevel gears and mountings are located in



FOR  $n$  ELEMENTS IN PARALLEL:  
 SYSTEM RELIABILITY =  
 $1 - (\text{FAILURE PROBABILITY OF ONE ELEMENT})^n$

FIGURE 28. RELIABILITY THROUGH REDUNDANCY.

opposite sides of the gearboxes, spaced and separated as in the case of the shafts. Any set of gears must, in case of a jamming-type failure, be able to separate without causing more than a local failure to the support structure. This presupposes a detail design of the case which would permit this, and would also contain the debris. Such a design is believed practicable.

3. The combining gears would be constructed as a parallel dual-face gear set. A tested example of this is shown in Figure 29. The gear is a double helical unit made of two single gears bolted to a common flange which is inserted between them. The test resulted in the nearly complete destruction of the teeth in one gear. The other gear successfully carried load for some time after failure of the first, and maintained complete synchronization of the transmission drive. A development of this principle is believed to be particularly well suited to the dual drive system combiner gear set.
4. The final drive is a planetary gear set comparable to that of Configuration I. The reasons for not duplicating the final drive are as follows:
  - a. Planetary gearing is inherently partially redundant by virtue of multiple meshing. This attribute may be developed, to increase the chances of survival after a failure, by various approaches now under consideration.
  - b. Historically, the higher-speed drive system elements are more prone to catastrophic-type failure. These are duplicated.
  - c. The mathematics of failure probability show that duplication of some elements in a connected series contributes to an overall improvement in the reliability of the series. Therefore, those elements that can be duplicated without incurring weight penalties are duplicated.

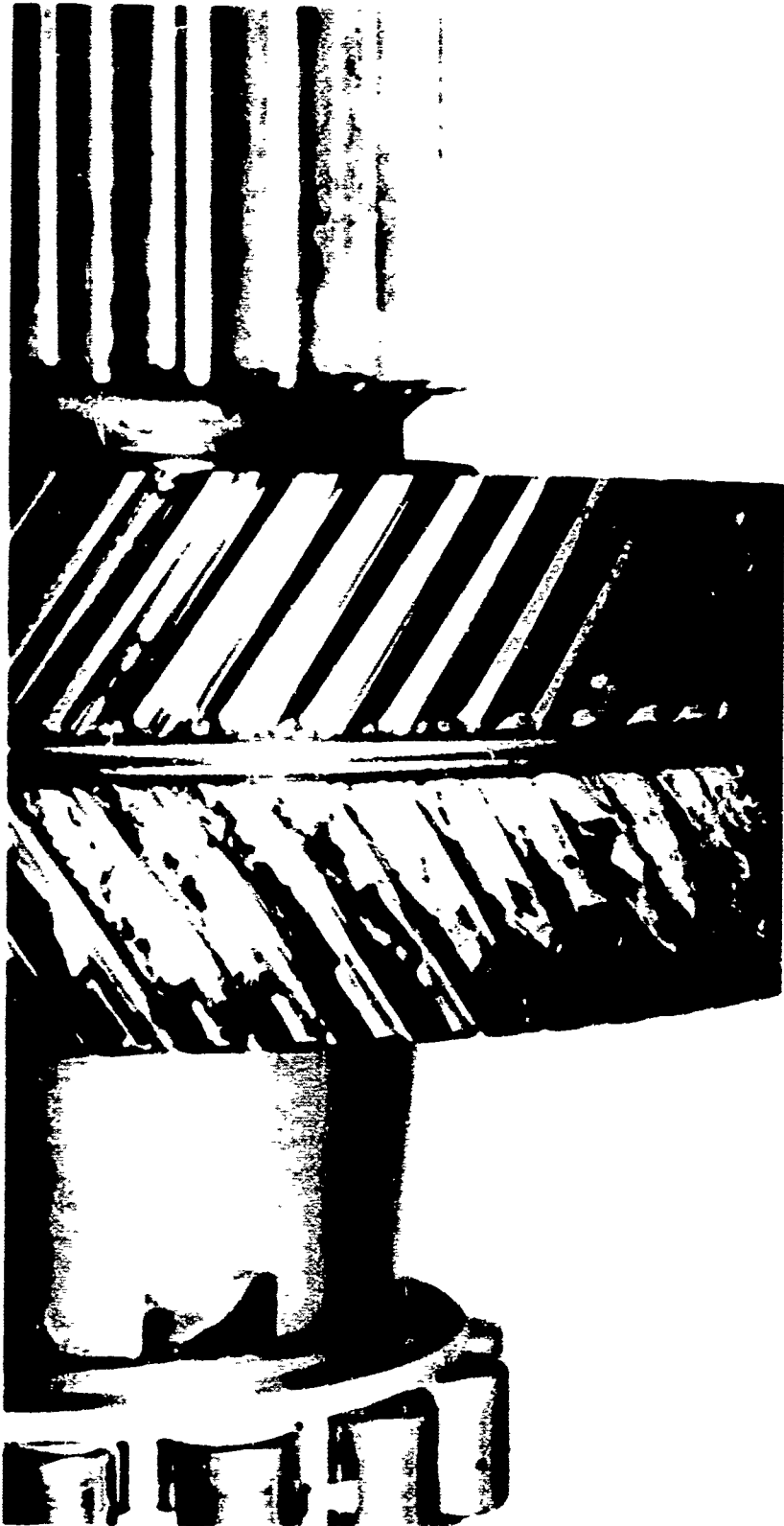


FIGURE 29. DUAL LOAD-PATH GEAR.

5. The estimated dead weight for Configuration III is close to that for single-shaft systems (I and II). This is because advantage is taken of the weight-saving possibilities of dual shafting in this arrangement. For example, the straight-through drive from the two rear engines permits the use of engine-speed dual shafts and gearing. The gears that collect the dual inputs are also used for speed reduction. The total number of meshes and the final reduction ratio are equivalent to those of a lower speed system (Configuration I) while the advantages of a high speed system are also obtained. A dualized conversion of a single system (Figure 34) will be heavier than the single system.
6. The rearmost engines, with air intakes as shown in Figure 48, have no ram air recovery in forward flight. The consequent power loss, as compared to full ram recovery, is estimated at 5.2 percent. The transverse engines are able to recover part of the ram effect, giving an estimated loss of 1.8 percent. Loss of ram recovery is not directly comparable to transmission power loss, since heat rejection devices are not involved. The effect is to increase specific fuel consumption, generally by 0.6 percent for each 1.0-percent power loss. At the critical hover condition the effect of reduced power must of course be considered along with propulsion system efficiency in the final design. In hover, power losses are estimated at 1.0 to 1.2 percent.
7. Dual shaft systems are advantageous to the HLH because power carried per bevel mesh is reduced to one-half that of the single-shaft system. The result is that the size and peripheral speed of the bevel gearing are only slightly greater than those employed in current systems.

## EVALUATION OF THE STUDY

### COMMITMENTS

The commitments of this study were several in number: (1) to define the high risk development areas of a tandem-rotor HLH mechanical drive system; (2) to indicate the exploratory development programs necessary prior to the final configuration design; (3) to estimate weight, size, and efficiency of the drive system.

To fulfill these commitments, the study investigated a number of mechanical transmission systems. All contain components which are possible candidates for inclusion in an HLH drive system; these systems were analyzed so that the study conclusions could be drawn from the broadest base. It was believed that concentration on one configuration only could obscure the fact that much configuration analysis and appraisal of the HLH itself remains to be done. This further and continuing effort may initiate requirements that would enhance the value of a currently less attractive transmission system.

A review of this rather wide selection of configurations showed that, in general, the transmissions do not require radical departures from present usage, either in principle or in physical size, to perform the HLH mission requirements. This is true, even though the horsepower transmitted is 3 to 4 times greater than that of present tandem-rotor machines. The reasons can be summarized as follows:

1. The tandem-rotor helicopter, by virtue of power sharing between two rotors, does not require in any one transmission the capacity required in that of a single-rotor helicopter of equivalent gross weight. Therefore, a tandem-rotor HLH does not require a major step in transmission size beyond present day single-rotor machines of the largest capacity.
2. Generally, transmission component diameters follow a cubic rule. Therefore, an increase of perhaps 200 percent in rotor power from a medium transport type single-rotor machine to an HLH tandem will increase sizes on the order of 25 percent.

3. Perhaps more important than size, component velocity and concomitant dynamic considerations can be held to a smaller increase over present practice than even 25 percent. This can be done by selection of gear ratios at the high-speed end, and by the lower rotor speed required by the rotor diameter necessary for the maximum gross weight specified.
4. Finally, techniques of dynamic control and analysis have progressed beyond those used in design of the current generation of helicopters. It is hoped that an aggressive program of still further basic improvement will be stimulated by customer interest. With the tools on hand, only evolutionary development in certain areas is necessary to assure high confidence in the HLH drive system. With successful improvement programs, it is believed that gains in weight and reliability will accrue to the benefit of the HLH.

The exploratory development programs found necessary to provide confidence in the HLH drive system are grouped below under the heading of PROBLEM-AREA PROGRAMS. In addition, under the heading of DEVELOPMENT-AREA PROGRAMS are grouped programs which, although applicable to helicopter drive systems generally, are expected to result in measurable benefit to the projected HLH. As such, they may be considered outside the primary obligation of this study. However, it is through these programs that the fullest realization of the HLH payload potential and overall effectiveness will be obtained.

#### RECOMMENDED PROBLEM-AREA PROGRAMS

The criteria used for defining a problem area are as follows:

1. The characteristics imposed by the requirements of the HLH are significantly more severe than those existing in current practice.
2. Analysis of the problem indicates dynamic effects that prohibit a confident extension of present practice without trial.
3. Solution is not necessarily possible in a short time span.

#### 4. Alternate approaches impose a penalty.

Items to which all of these criteria apply are considered mandatory development items. They apply specifically to the HLH.

#### Overrunning Clutches

Location of the overrunning clutch on the transmission input shaft (Figure 40) results in a clutch rubbing velocity 50-percent higher than current experience. It also places the clutch in close proximity to turbine-frequency vibration sources. Power transmittal requirements also contribute to the increased velocity. The recommended approaches to increasing confidence in overrunning clutches applied to the HLH would include an investigation of the factors which influence the ability of sprag-type clutches to survive a high rubbing velocity. Material hardness and finish, sprag geometry, and lubrication would be considered. An investigation of the hydrodynamic lifting of sprags from the rotating race should be made. This might be conducted by methods similar to those used in determining gear tooth oil-film thickness.

A first-phase effort should include overrunning tests of present-day-technology sprag clutches, with variations in lubrication as to quantity, points of admission to, and egress from, the working surface. It is understood that such testing will be conducted in the near future by a major clutch manufacturer. An investigation must also be made of the vibratory impulses which may be imposed by location of the clutch at the engine-transmission interface; additional inputs from the damped, supercritical speed shafting should be included.

It is also recommended that a study be made of the overrunning clutch function. This would determine whether fundamental changes in design approach can reduce sensitivity to lubrication, improve reengagement control, and increase overall reliability. This study should also consider elimination of the mechanical overrunning clutch, possibly by transferring some of its functions to the power turbine.

#### Scoring of Gear Teeth

The unusually large loads imposed upon the HLH gearing indicate

the need for coarse gear tooth pitches, in order to realize a minimum-weight gear set with a satisfactory balance between strength and surface durability. Coarse pitches, combined with other factors which may be present, are conducive to early scoring-type failure. It is therefore recommended that further investigation be conducted to increase the confidence in these gear applications. This would include the following:

1. Detail examination of typical HLH pitches for scoring, using advanced computer techniques for optimization of the form.
2. Tests of sample gears using optimized tooth forms, various surface finishes, and various materials and surface treatments.
3. Verification that the analytical technique provides design information sufficient to eliminate scoring in HLH-type gears.

#### Engine Control System for a Four-Engined Helicopter

Current twin-engined helicopters present a number of power management problems to the pilot. Maintaining uniform torque output from each engine may divert the pilot's attention from primary flight functions. For this reason, automatic power management is being developed for turboshaft engines to be used in twin-engined helicopters. The extension of this principle to four-engined helicopters should be investigated.

Since the engines presently being considered for the HLH do not have an automatic power management system, the problem of power management for a four-engined helicopter must be investigated to define its scope.

A program for setting up and operating a four-engine simulator with engine instrumentation is recommended. The results of engine and flight-control inputs would be presented through the instrumentation, thereby permitting evaluation of the problem of pilot power management.

#### RECOMMENDED DEVELOPMENT-AREA PROGRAM

It is highly desirable that certain technical areas be given

the stimulus of development, in order to provide advantages for all classes of helicopters equipped with mechanical drive systems. To illustrate the effect of advanced stress levels upon the HLH transmission, design studies are shown and are summarized in Figure 7 and Figure 8. Attainment of these stress levels is believed practicable in the 1970 time period; however, this will necessitate pursuit of active development programs in the immediate future.

The recommended development areas which have shown improvement potential in the design study are as follows:

#### Improvement in Gear Tooth Strength

A demonstrated improvement in bending fatigue strength of gear teeth will reduce tooth size and tend to reduce both scoring and dynamic problems. The improvement is also desirable to provide reductions in weight and size.

The recommended program would include the following:

1. Analysis of the geometry factors influencing gear tooth strength
2. A continuation of the gear fatigue testing conducted by the Vertol Division, with the objective of improving processing variables such as heat treatment, and of investigating improved materials and gear tooth forms
3. Sufficient testing to provide a high level of confidence in the reliability of gearing at the increased stress levels, with optimized combinations of form, material, and process technique

#### Supercritical-Speed Shafting

The supercritical speed shaft has proved of particular advantage to the HLH configurations under study. With subcritical speed shafting, the combination of increased shaft length and increased horsepower required for the HLH would make for increases in shaft weight and complexity. However, by substituting supercritical speed shafting, major gains are realized in several areas. These are: (1) an estimated 30-percent reduction in shaft weight, and (2) a 75-percent reduction in the number of major component assemblies including supports. Each

extra support assembly represents additional mechanical complexity and is, therefore, a potential failure origin, as well as being another maintenance item.

A recommended program for the further development of the super-critical speed shaft would be directed as follows:

1. Continuation at Vertol Division of present bench tests of a full-scale shaft, culminating in testing of the shaft installed in an aircraft
2. Development of an aircraft-type damper suitable for use on the HLH shaft system

### Bearings

In addition to the general development effort being exerted by the bearing manufacturers to improve materials and manufacturing processes, it is necessary that specialized studies be conducted by the transmission designer. These will lead to further understanding of specific helicopter transmission problems. It is therefore recommended that the following investigations be pursued:

#### Planet Support Bearing

To achieve the reduced planetary diameter made possible by full use of improved bearing capacity, it is desirable to extend existing analytical techniques. The type of loading, outer race deflection, and centrifugal effect are all unique to a planet bearing. The planet gear surrounds the bearing as a rotating outer race. Loads are applied to the flexible (outer) race at opposite points.

The resultant deflections have been shown to have substantial effect on the position of the load zone, and upon peak loads. Preliminary analysis (Reference 2) has indicated the benefits of permanently deforming the inner race to compensate for elastic deformation of the outer. This would be designed to extend the load zone and average peak loads.

A computer program should be formulated to further analyze the effect on roller loads. A follow-on test program is

necessary to evaluate the analysis, using an existing planetary system.

#### High-Speed Bearings

Again, the confident use of improved bearing capacity makes necessary a reduction in simplifying assumptions. Presently available programs are progressing toward a rigorous analysis of the rolling-element forces and motions which determine actual bearing life. It is necessary to continue and systematize this analysis, to define more exactly these body forces. It is also important to investigate the circumstances that cause bearing elements to skid instead of roll, and to eliminate this problem.

The extension of present analytical effort to include body forces arising from high speed operation is required. The effective spring rate of the rotating bearing should also be qualified, to better solve for bearing external loads. The effect of misalignment on roller bearings, with resulting edge loading of the rollers, should be analyzed, and verified by test.

#### Bearing Cage Design

In recent years, aircraft bearing capacity has been raised by increasing the size and number of rolling elements. This has, within a given envelope, reduced the space available for a cage to locate and control the elements. So far as is known, this increasingly-critical component has not received analytical treatment to determine the forces acting upon it, and the consequent stress levels. Bearing cage design should be reviewed, following analytical effort, and various designs, materials, dimensional clearances, and lubricant coatings evaluated by test programs. This work will be directly applicable to increasing the emergency nonlubricated capability of bearings, since the cage is considered the most critical bearing element when lubrication is denied.

#### Low-Speed Bearings

A bearing, such as a rotor shaft support bearing, operating at low speed and under high load is subject to unique problems. It is postulated that certain speed-load combin-

ations prevent formation of a sufficient oil film to separate the race and rolling elements, thereby leading to premature failure. Advanced bearing materials may reduce this problem, but the effect is substantially to derate the bearing, by using up potential capacity improvement to compensate for a problem. Successful investigation and solution will allow significant weight saving in these normally large and heavy bearings.

A program to investigate lubrication should be initiated. This would include application of lubricating oil to different bearing areas, and also the combination of fluid and solid-film lubrication. An extended program would also investigate specialized types of bearings fundamentally designed to withstand the combined radial and thrust load conditions typical of these applications.

#### Planet Load Equalization

The final reduction planetary system accounts for two-thirds of the gearing weight of the drive system. It is, therefore, the area where substantial weight gain can be realized from successful improvement techniques. The maldistribution of planet loading increases planet gear and bearing stresses, which in turn increase the planet system weight. Equalization of planet load can reduce the cyclic load peaks, enabling higher power to be transmitted through equivalent weight gears. New approaches are required, to provide load equalization without increasing complexity and weight. A study of design parameters is recommended, together with a comparison of existing test information with calculation. If a satisfactory solution is found, it would be recommended that the program be continued to prototype evaluation, using existing transmission and stands as test hardware.

#### WEIGHT EVALUATION

To obtain a realistic weight estimate of a drive system from conceptual drawings requires the application of experience factors obtained from designing and building helicopter drive systems. Without this background, the estimation is almost certain to neglect hidden factors which significantly influence system weight. Trend curves were used to cross-check calculation in the transmission weight estimates of this report.

In reviewing the drive system weights of the six systems studied in this report, several important points warrant discussion. These points are:

1. Dead weight comparison between the six systems
2. Total effective weight (including power loss) and its effect on comparison of systems
3. The effect of designing for available power
4. The correlation in major subsystem weights between the HLH and current experience
5. The weight effect of transmission improvements

#### Dead Weight Considerations

The six study drive systems, compared on a dead-weight basis (neglecting power loss), show a difference of 7 percent between the lightest and the heaviest (Figure 30). This is not entirely unexpected, since these configurations were selected from a large number of possible candidates as being likely to produce minimum-weight systems. Compensating features among the systems tend to equalize weight. The additional weight of the engine nose gearboxes (Configuration IV), for example, was counterbalanced by a somewhat lighter combining transmission than is found in Configuration I. The weight advantage of higher shaft rpm was cancelled by extra weight in the reduction gearing (Configuration I alternate). The weight of additional bevel gearing was compensated for by the higher speed and lower torque made possible, without additional meshes, by the position of the combining gears (Configuration III, Dual Shaft System). Tables V through VIII show subsystem weights. Figure 31 compares historical trends with HLH system weights.

#### Total Effective Weight Considerations

The effect of power loss, when added to dead weight, makes the comparison between systems more telling. There is a spread of approximately 13 percent between the highest and lowest total effective weight. Total effective weight is a better selection criterion than dead weight since it accounts for the power that is not available to lift, but which instead is turned to heat

DESIGN CONDITIONS  
 15,200 TOTAL SHP  
 134 ROTOR RPM  
 ADVANCE STRESS LEVELS

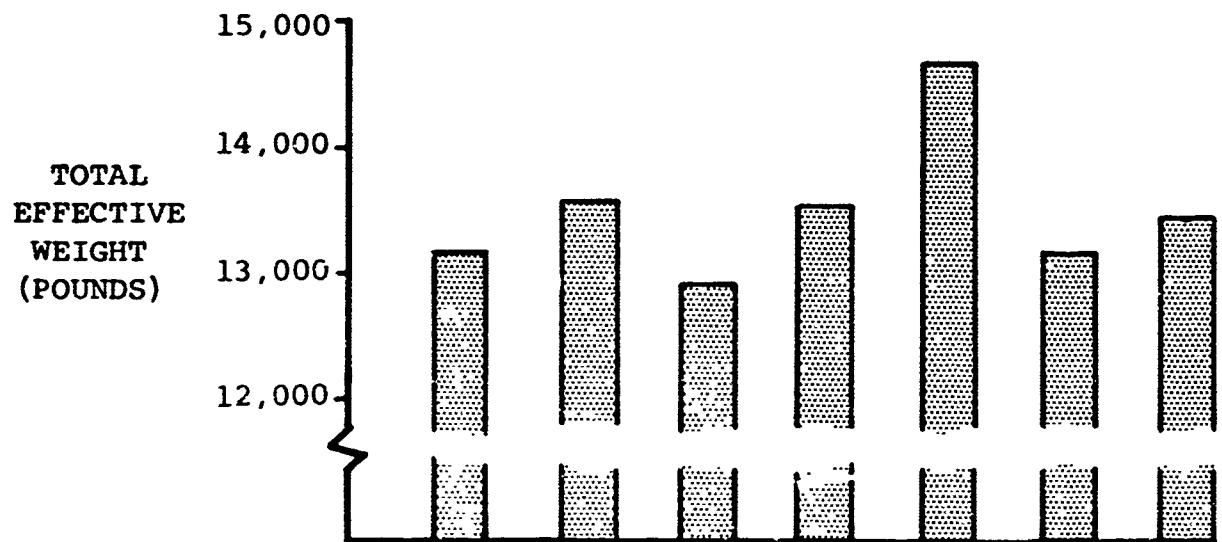
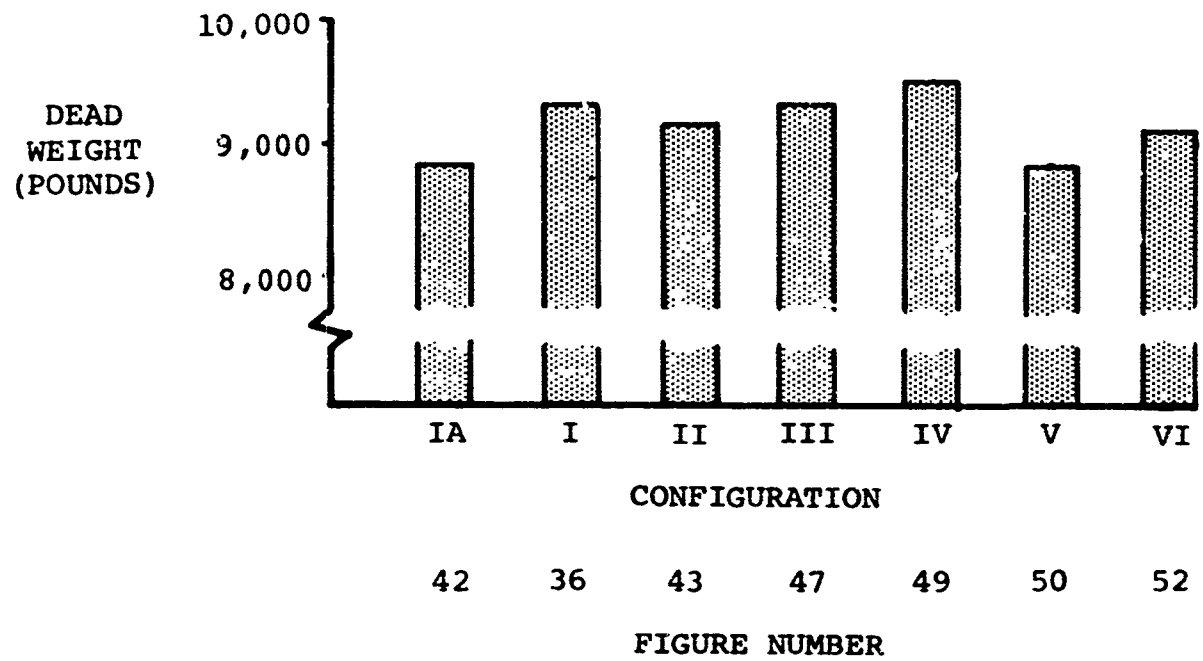


FIGURE 30. WEIGHT COMPARISON OF STUDY DRIVE SYSTEMS.

TABLE V. CONFIGURATION IA HELICOPTER WEIGHT SUMMARY  
(TRANSPORT FUSELAGE)

Item		Mission Weight (lb)		
		Transport	Heavy- Lift	Ferry (1800 nm)
Weight empty:		44,576	44,576	44,576
Rotor group (Note 1)	9,520			
Body group	10,380			
Alighting gear	3,940			
Flight controls	2,706			
Propulsion group (Note 2)	12,213			
Electrical and electronics	1,275			
Furnishings and equipment (Note 3)	824			
Miscellaneous (Note 4)	3,718			
	44,576			
Fixed useful load		890	890	890
Fuel		8,550	3,850	45,034
Auxiliary tankage		---	---	5,500
Cargo		24,000	40,000	--
Takeoff gross weight		78,016	89,016	96,000
Design gross weight		80,000	---	---

Notes

1. Three rotor blades with 48-foot radius and 3.17-foot chord.
2. Transmission design limit is 15,200 shaft horsepower.
3. Cabin interior includes cargo provision for troops.
4. Includes 5-winch cargo system.

TABLE VI. CONFIGURATION IA PROPULSION GROUP WEIGHT SUMMARY

Item	Weight (lb)
Engines - LTC 4K-2 (4 req'd)	2,500
Air induction (4 req'd)	20
Exhaust (4 req'd)	64
Lube system - engine (4 req'd)	30
Fuel system:	480
Tanks, fittings	345
Engine lines	60
Boost pumps	12
Systems, controls, supports	63
	<u>480</u>
Engine controls (4 req'd)	92
Starting - hydraulic - engine (4 req'd)	178
Drive system:	8,849
Aft transmission	4,080
Forward transmission	3,306
Aft rotor shaft	320
Interconnect shaft	327
Engine shafts	50
Lube system	625
Rotor brake	111
Miscellaneous	30
	<u>8,849</u>
Total	<u>12,213</u>

TABLE VII. CONFIGURATION IA LUBRICATION SYSTEM WEIGHT SUMMARY

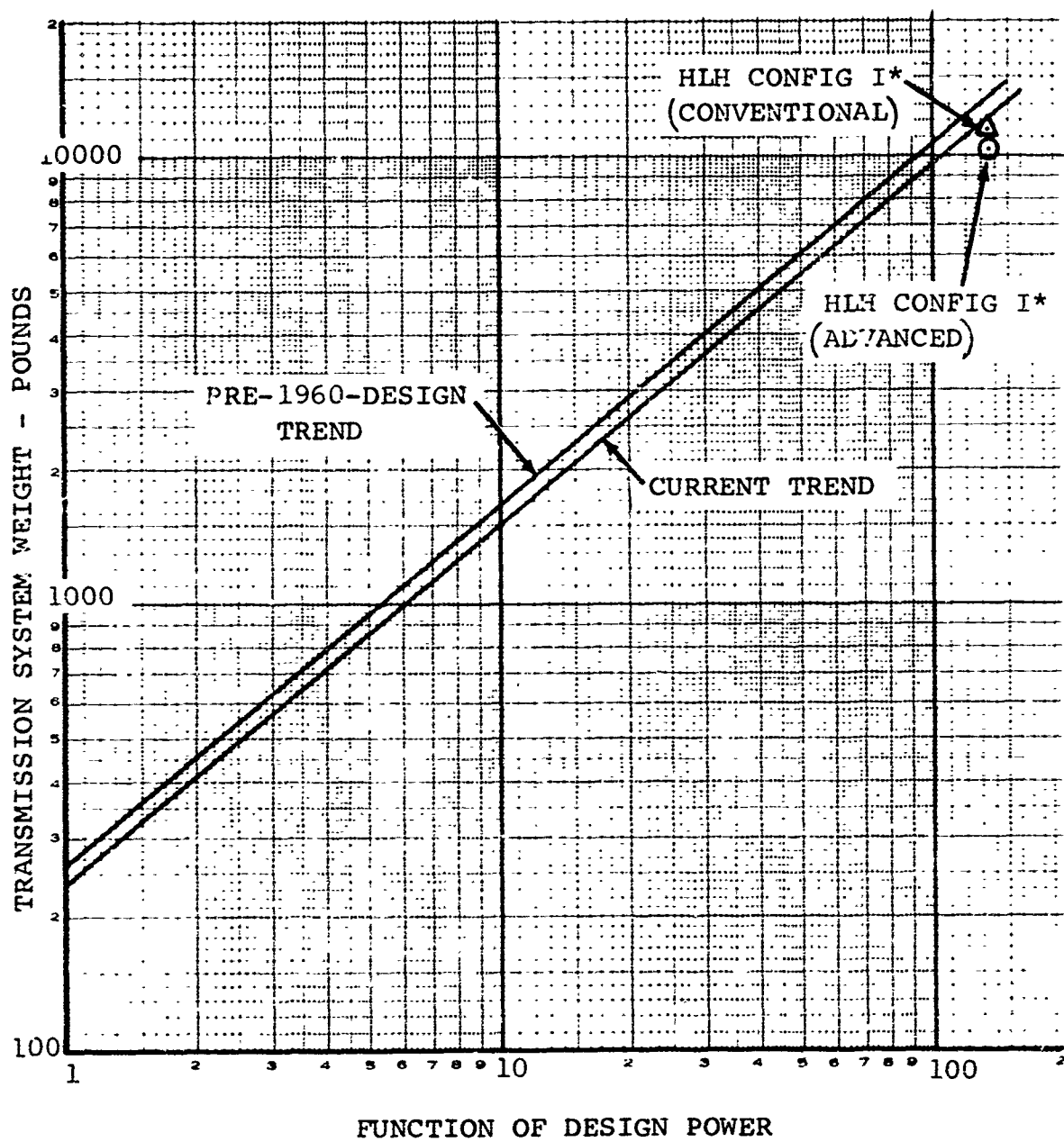
Item	Weight (lb)
Oil cooler - fwd transmission	54
Oil cooler - aft transmission	87
Blower and drive shaft	55
Plumbing	181
Supports	11
Cooler baffles	5
Oil (at 7.75 lb per gallon)	<u>232</u>
Total	<u>625</u>

TABLE VIII. COMPONENT WEIGHT SUMMARY

Component	Weight (lb) for Configurations					
	I	II	III	IV	V	VI
Aft transmission (Note 1)	2886	2511	2734	2371	4220	2844
Fwd transmission (Note 2)	3306	3732	3539	3304	3304	3377
Engine transmission	--	--	--	472	--	--
Combining transmission	--	--	--	352	--	--
Aft rotor shaft (Note 3)	1946	1946	1946	1946	320	1946
Interconnect and engine shaft	377	269	284	377	367	377
Lube system (Note 4)	595	575	625	655	625	595
Rotor brake	111	111	111	111	111	111
Miscellaneous	30	30	30	30	30	30
Total	9251	9174	9269	9507	8866	9169

Notes:

1. Aft transmission includes accessory drive and blower drive; combining section where applicable.
2. Fwd transmission includes fwd rotor shaft.
3. Aft rotor shaft includes thrust bearing.
4. Lube system includes oil.



\*AT 15,200 HP

FIGURE 31. WEIGHT TRENDS COMPARISON.

and dissipated with resulting additional weight for coolers and blower. Configuration II, because of the elimination of power carry-through in the interconnect shaft bevel gearing, has a low number of meshes, and therefore, a low total effective weight. Configurations IA, I, III, V, and VI carry slightly higher total effective weight; Configuration IV is substantially higher.

The conclusion to be drawn is that selected best transmission systems can vary considerably in arrangement, and yet be identical, within the limits of accuracy of trend calculation, in dead weight.

The selection of the optimum arrangement was, in this case, based upon factors other than dead weight.

#### Effect of Power Criteria

The drive system power criteria used in this study were based upon the transmittal of full available power from advanced versions of current engines, at a rotor rpm corresponding to a 48-foot blade radius. The system weights (Figure 30) uniformly reflect these criteria. Less severe criteria could be considered--for example, the power required to meet exactly the mission specification (11,500 shp). The effect on transmission system weight is shown in Figure 32. This shows that, by limiting power to 11,500 shp, a weight reduction of 20 percent should be realized, resulting in a Configuration I system weight of 7100 pounds.

It was believed that the commitments of this study were best met by using transmission design criteria which would be consistent with potential growth horsepower. It is historically demonstrable that powerplant growth leads transmission capability and that, as increased power becomes available, improved helicopter performance is readily accepted. Transmission problem areas have therefore been defined for a system of sufficient capacity to accept this growth power, with attendant component size, velocity, and weight. It is emphasized that final design of the HLH drive system will reconsider alternate power criteria and that, in consequence, the weight of the system may be reduced.

CONFIGURATION IA SYSTEM  
ADVANCED STRESS LEVELS EMPLOYED  
ROTOR RPM IS CONSTANT (134 RPM)  
ROTOR DIA IS CONSTANT ( 96 FT)

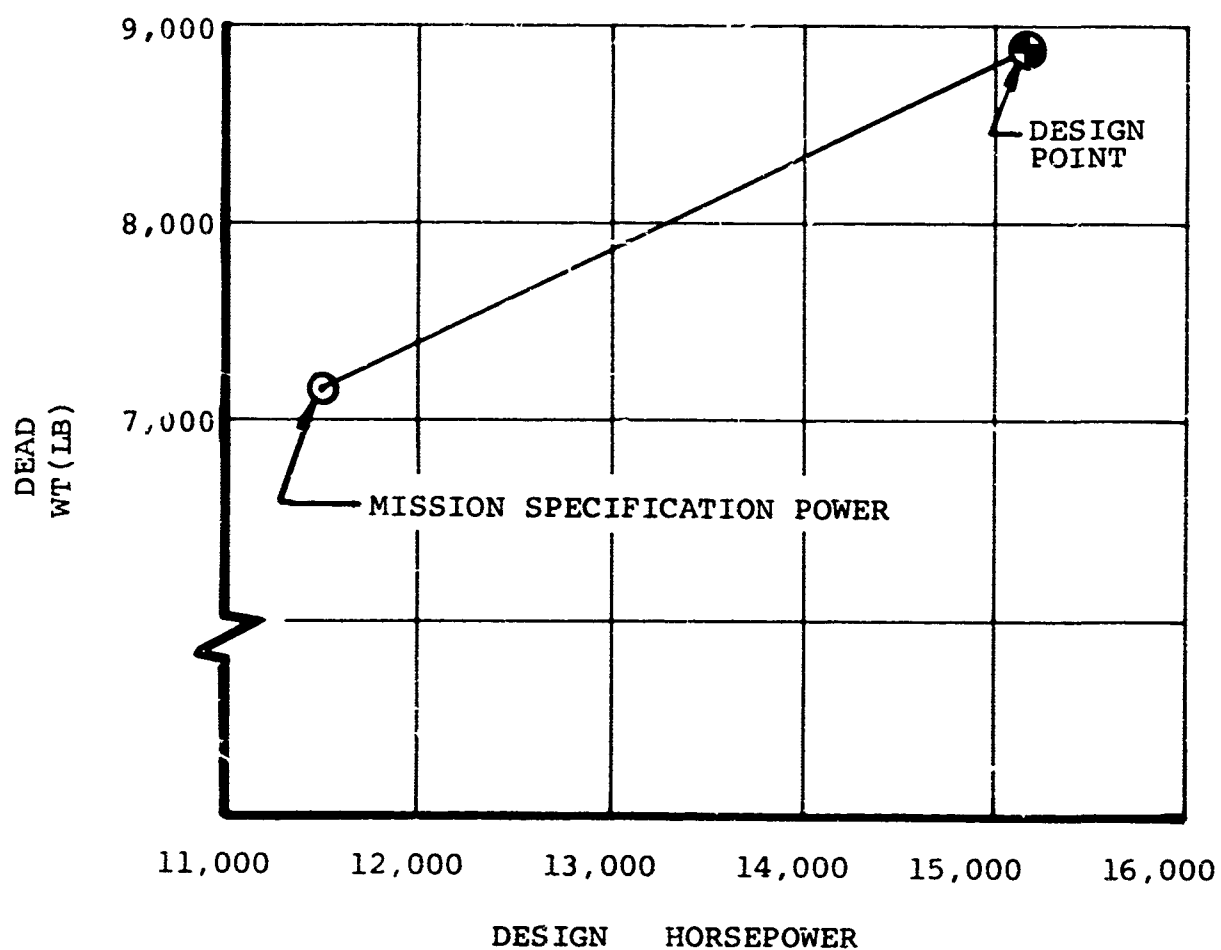


FIGURE 32. EFFECT OF DESIGN HORSEPOWER ON WEIGHT.

### Correlation of HLH to Current Tandem-Helicopter Transmission Weights

Subsystem weight percentages showed close agreement with those currently experienced in tandem-rotor drive systems. The major difference was in the percent system weight attributed to the aft rotor shaft. Rotor blade radius is an influencing factor on pylon height; pylon height determines the necessary length of this shaft. The blade radius selected for this study was sufficient to increase the aft shaft portion of the total weight. An approach to reducing shaft weight is to place the final reduction immediately beneath the aft rotor. A vertical interconnect shaft replaces the rotor shaft from bevel case to final reduction; weight is reduced as a consequence of the reduced torque. Bending forces and bending stiffness requirements are eliminated by a moment-carrying support for the rotor shaft above the final reduction gearing.

### Effect of Transmission Improvements

The transmission improvement areas which contribute to weight reduction and which are embodied in the estimates (Figure 30) are advanced gear stress levels, increased bearing capacity, and supercritical speed shafting. The combined effect of these improvements is to give an estimated 1200-pound weight reduction in Configuration I drive system, as compared to the same configuration using 1960-era design practice. This represents a 13-percent weight reduction, at no sacrifice in transmission capacity.

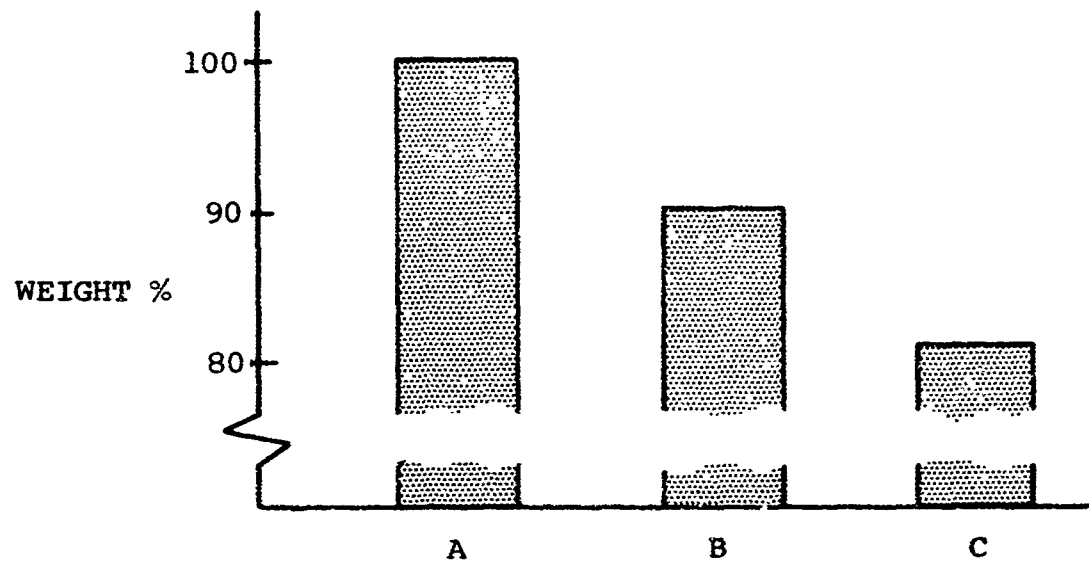
The magnitude of stress-level increase and the effect of supercritical shafting is detailed in foregoing sections. The attainment of these improvements is possible within the time period allotted for HLH development. To meet overall reliability and maintainability goals, it will be necessary to pursue development programs such as those outlined in the preceding section. Existing programs leading to the realization of operational supercritical shafting and improved bearings must be continued.

Decreased face width and diameter of planetary and bevel gearing account for the largest part of the reduction in system weight. Increased bearing capacity makes possible a further reduction, especially in the planetary section. Reduced gear

diameters impose greater reaction loads because of higher tangential forces. Without a commensurate improvement in bearing capacity, it is found that bearing diameter often becomes the decisive sizing factor for housings, mountings, and planet ring gears.

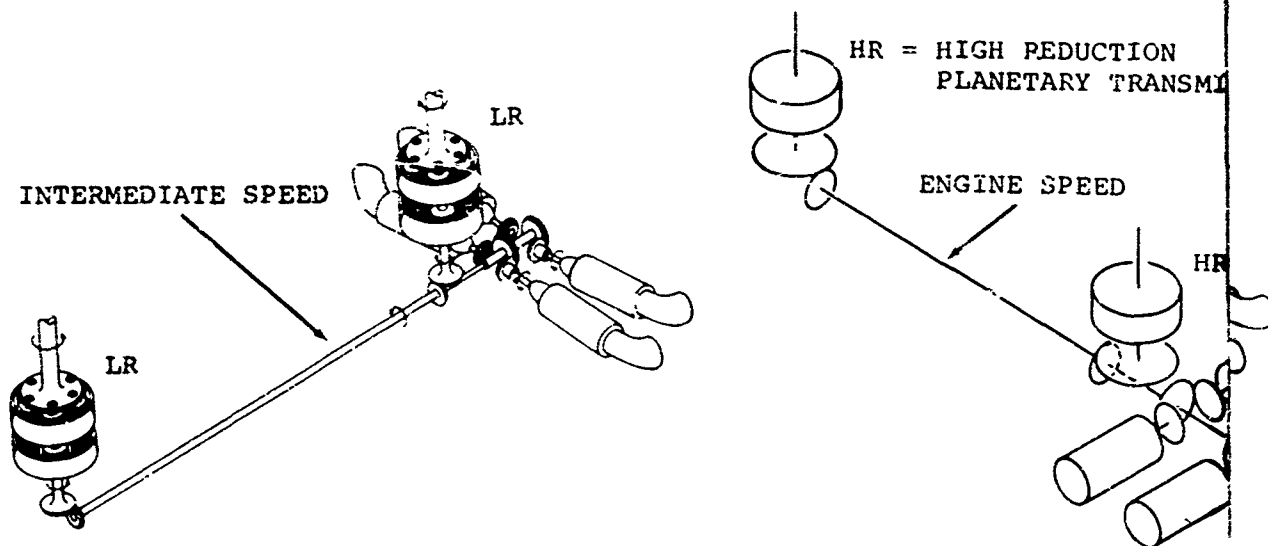
An additional approach to weight reduction is the substitution of titanium for ferrous components other than gears and bearings. These could include rotor shafts, gear shafts and carriers, shaft adapters and couplings, bolting hardware, spacers, and shims. The decision to use titanium for a specific application is predicated upon continuing research on the material characteristics and allowable stress levels. Such a program is presently under way at Vertol Division. It is also predicated upon stiffness requirements, especially in shaft applications, where the lowered modulus of titanium may require increased diameter to maintain stiffness while preserving the weight advantage.

Assuming that maximum use can be made of titanium in the above mentioned components, the weight advantage to the HLH drive system represents an estimated 800 pounds. This latter approach can be applied separately from that of advanced gear and bearing stress levels, since the two approaches are independent. Added together, they represent a possible drive system weight saving of 16 percent of a conventionally designed (1960-era) system. System weights shown in Figure 30 do not include the effects of using titanium. Figure 33 compares the weight effect of conventional design, first with that of advanced stress levels and supercritical shafting, and then with the effect of these plus titanium.

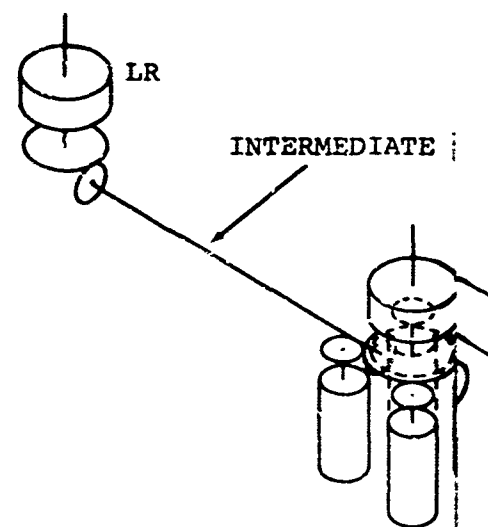
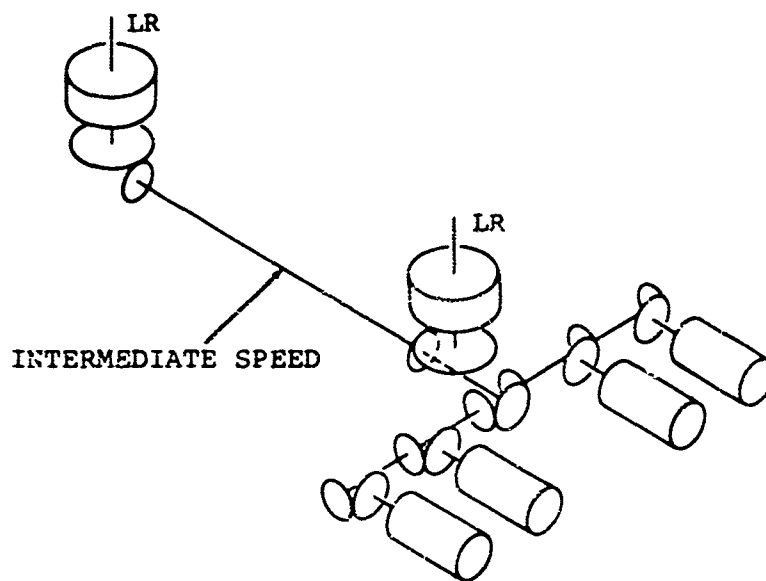


- A - CONVENTIONAL DESIGN AND STRESS LEVELS
- B - ADVANCED STRESS LEVELS, SUPERCRITICAL SHAFTING  
(SHOWN IN FIGURE 30)
- C - ADVANCED STRESS LEVELS, SUPERCRITICAL SHAFTING PLUS  
TITANIUM

FIGURE 33. WEIGHT EFFECT OF STRESS LEVEL AND MATERIAL.



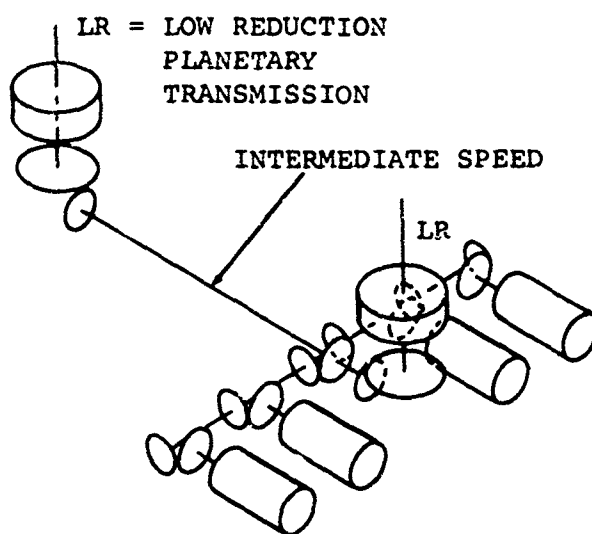
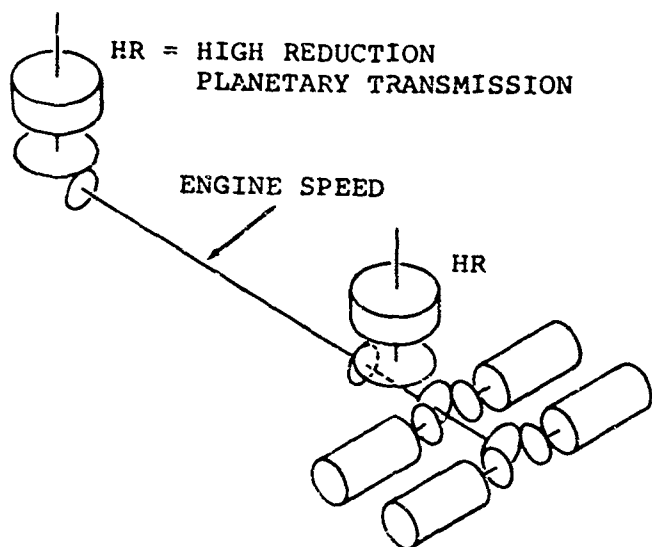
CONFIGURATION I			CONFIGURATION VII		
5.03	2	38 (10 S/B)	5.29	2	42 (10 S/B)
NO. EQUIV. MESHES PER ENG.	NO. XMSN BOXES	NO. GEARS AND (S/B)	NO. EQUIV. MESHES PER ENG.	NO. XMSN BOXES	NO. GEARS AND (S/B)
6.03	6	43 (15 S/B)	5.03	2	37 (4 S/B)



CONFIGURATION VIII

CONFIGURATION V

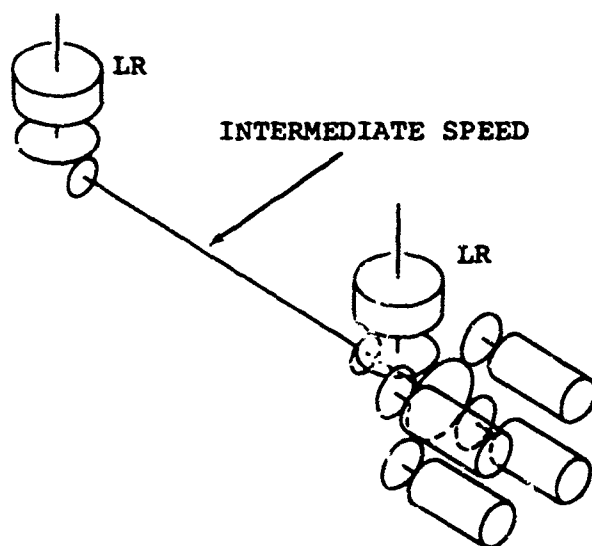
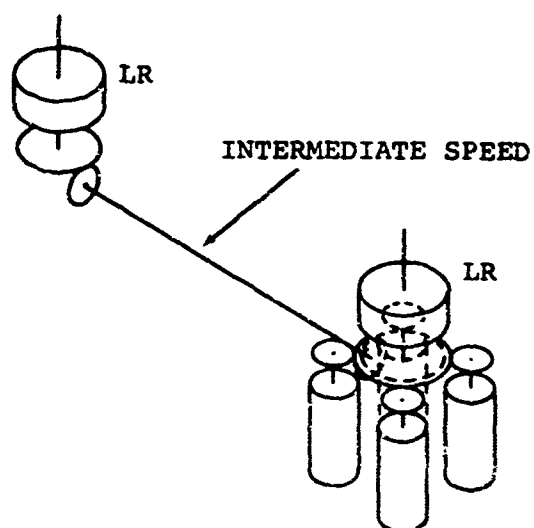
FIGURE 34. CANDIDATE CONFIGURATIONS



CONFIGURATION VII

CONFIGURATION IV

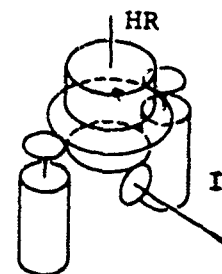
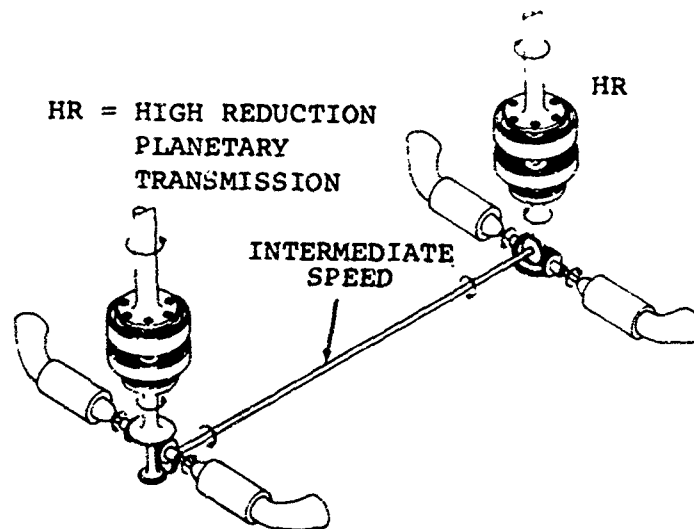
5.29	2	42 (10 S/B)	6.03	6	43 (15 S/B)
NO. EQUIV. MESHES PER ENG.	NO. XMSN BOXES	NO. GEARS AND (S/B)	NO. EQUIV. MESHES PER ENG.	NO. XMSN BOXES	NO. GEARS AND (S/B)
5.03	2	37 (4 S/B)	5.03	2	37 (4 S/B)



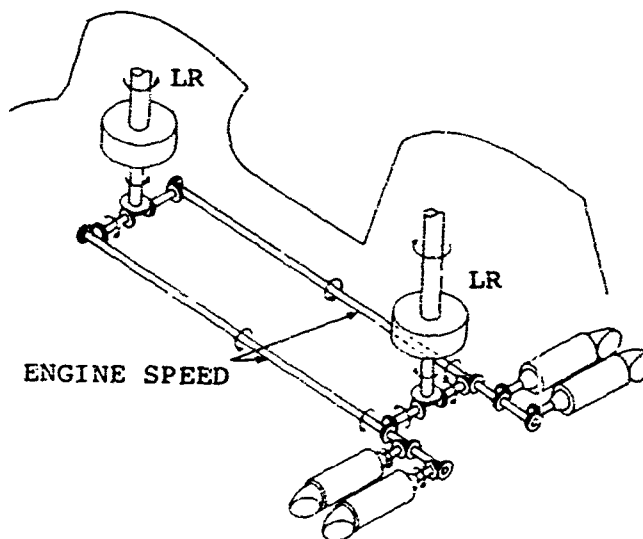
CONFIGURATION V

CONFIGURATION VI

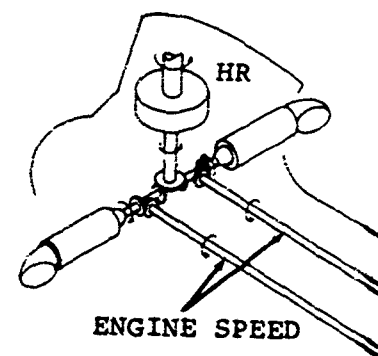
FIGURE 34. CANDIDATE CONFIGURATIONS. (Sheet 1 of 2)



CONFIGURATION II			CONFIGURAT:	
4.29	2	42 (10 S/B)	4.79	2
NO. EQUIV. MESHES PER ENG.	NO. XMSN BOXES	NO. GEARS AND (S/B)	NO. EQUIV. MESHES PER ENG.	NO. XMSN BOXES
6.03	2	50 (22 S/B)	4.29	2

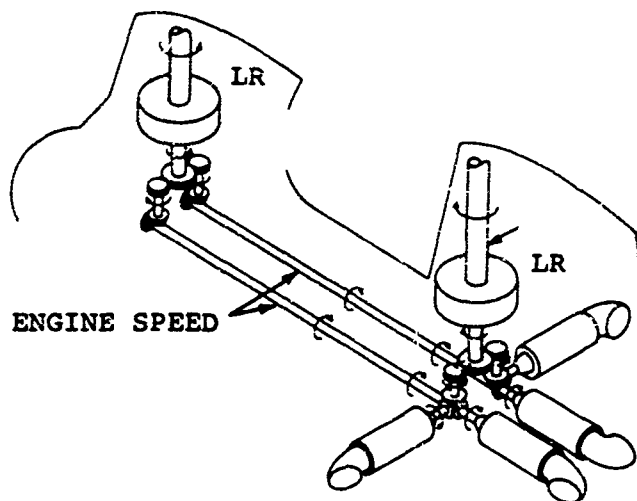
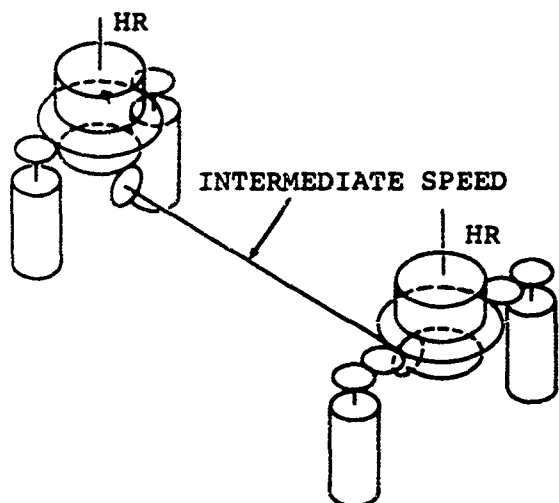


CONFIGURATION X

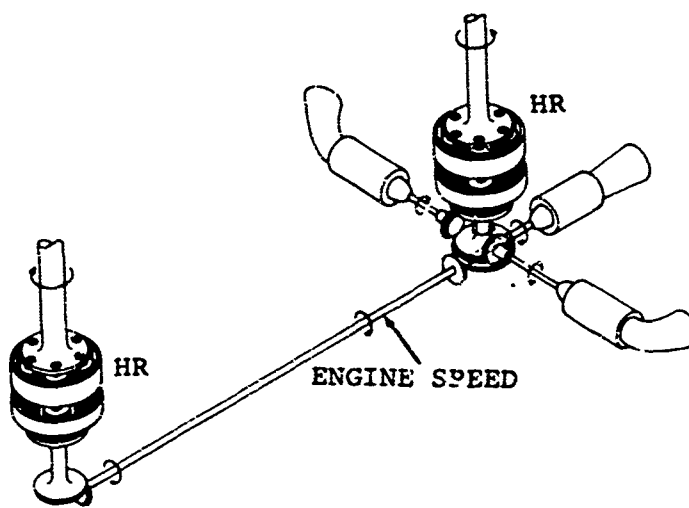
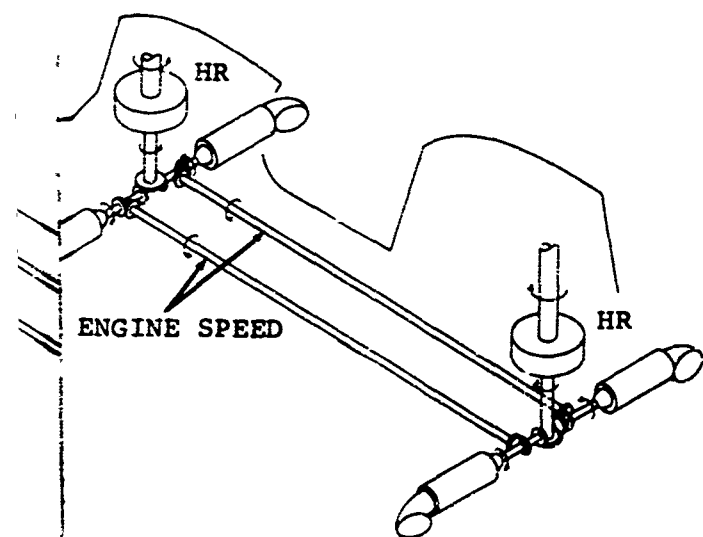


CONFIGURAT

FIGURE 34. CANDIDAT



AT	CONFIGURATION IX				CONFIGURATION III		
	4.79	2	44 (4 S/B)		5.03	2	44 (10 S/B)
ES	NO. EQUIV. MESHES PER ENG.	NO. XMSN BOXES	NO. GEARS AND (S/B)		NO. EQUIV. MESHES PER ENG.	NO. XMSN BOXES	NO. GEARS AND (S/B)
	4.29	2	46 (14 S/B)		4.62	2	39 (7 S/B)



AT: CONFIGURATION XI

CONFIGURATION XII

FIGURE 34. CANDIDATE CONFIGURATIONS. (Sheet 2 of 2)

**B**

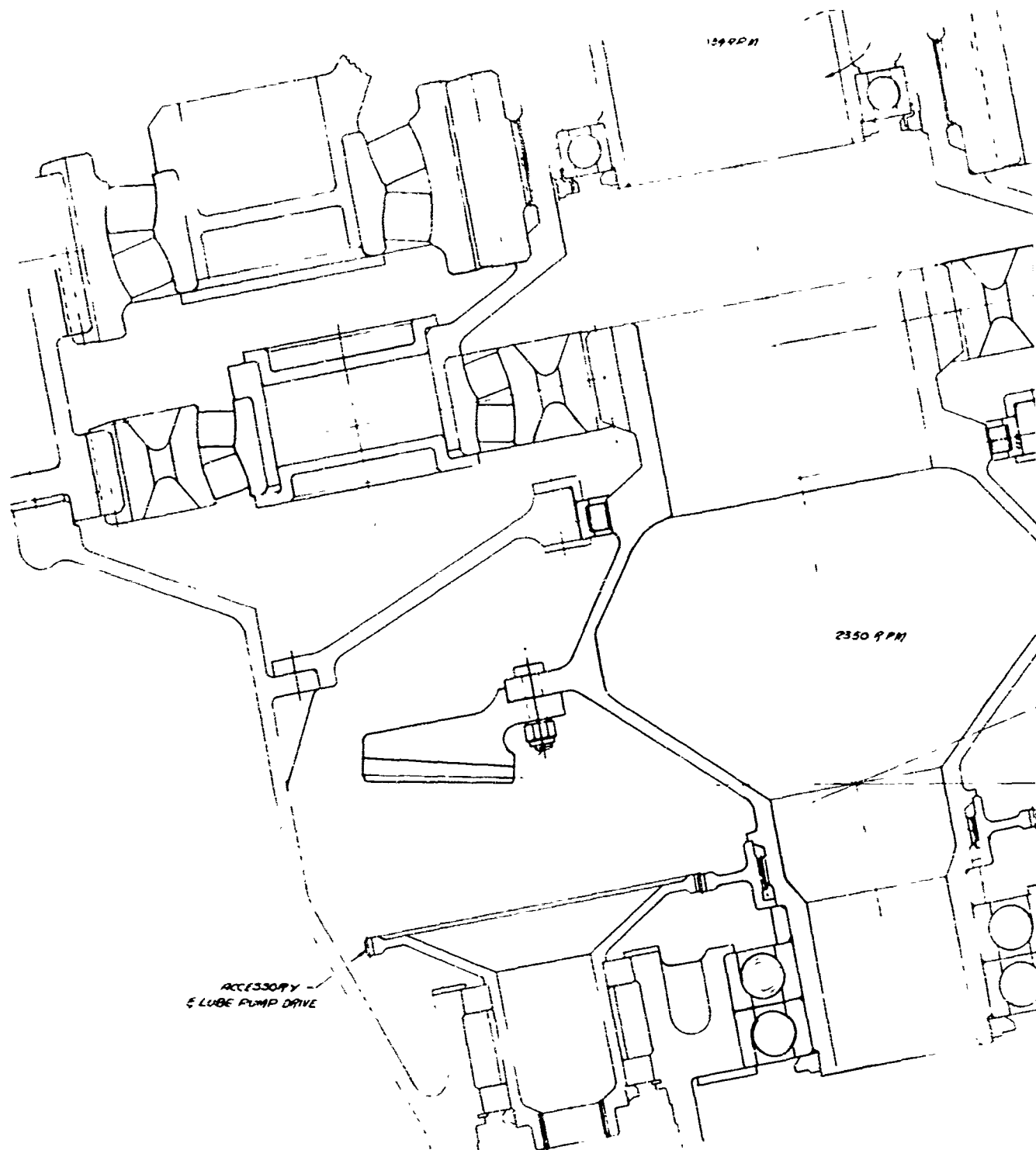
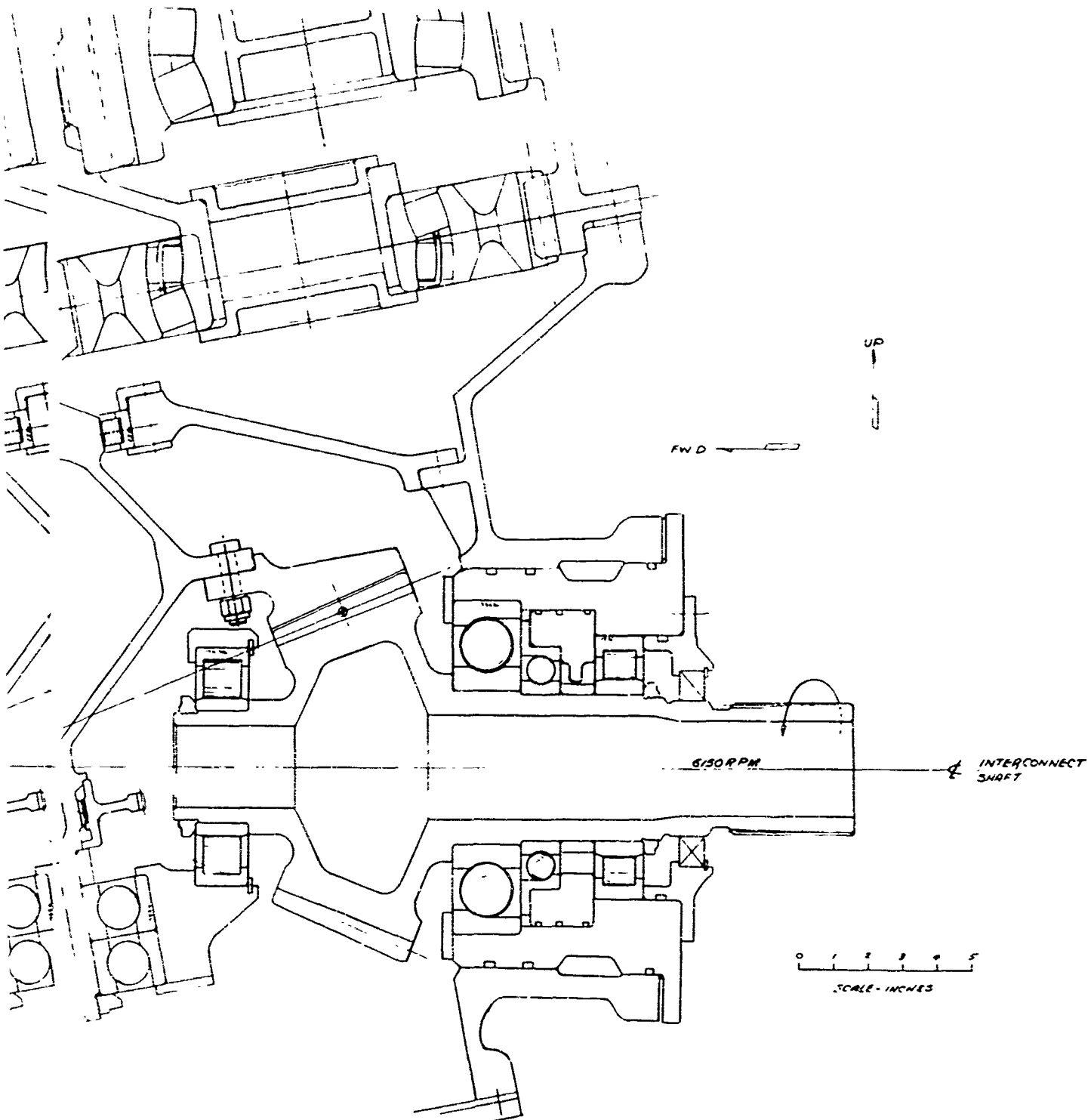


FIGURE 35. FORWARD ROTOR TRANSMISSION - CONVENTIONAL STRESS LEVEL GEARS AND BEARINGS.



**B**

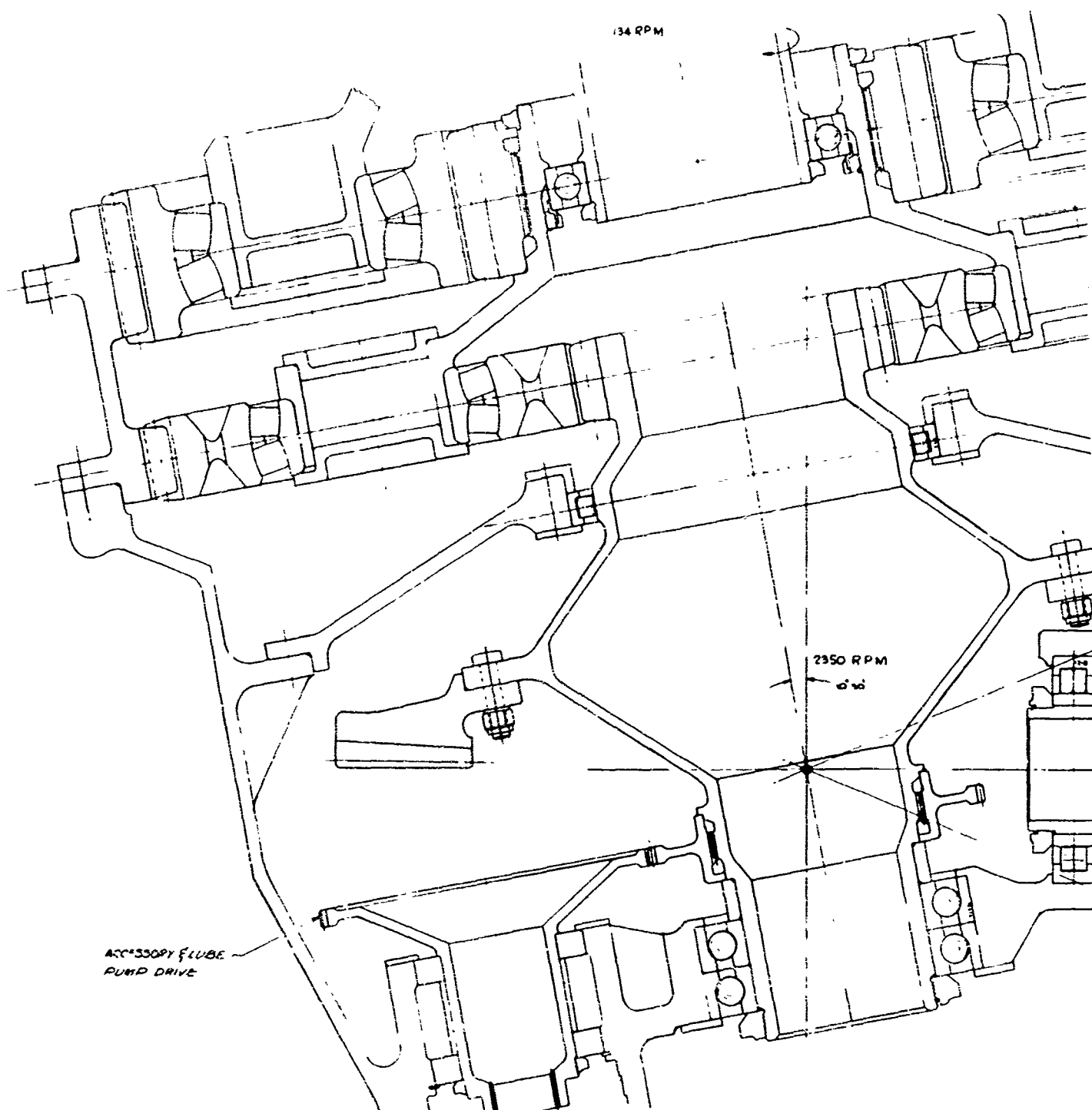
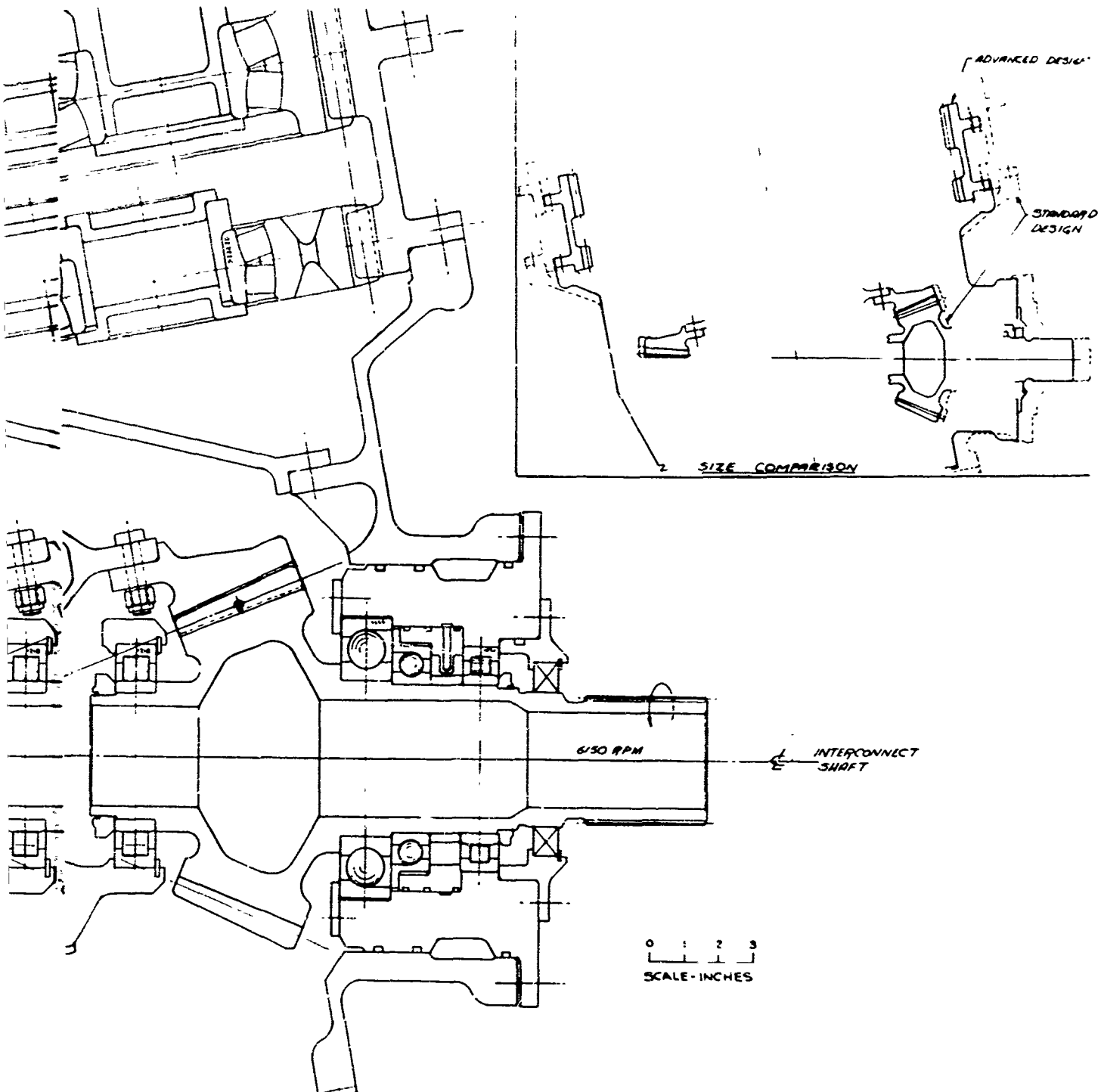


FIGURE 36. FORWARD ROTOR TRANSMISSION - ADVANCED STRESS LEVEL GEARS AND BEARINGS.



**B**

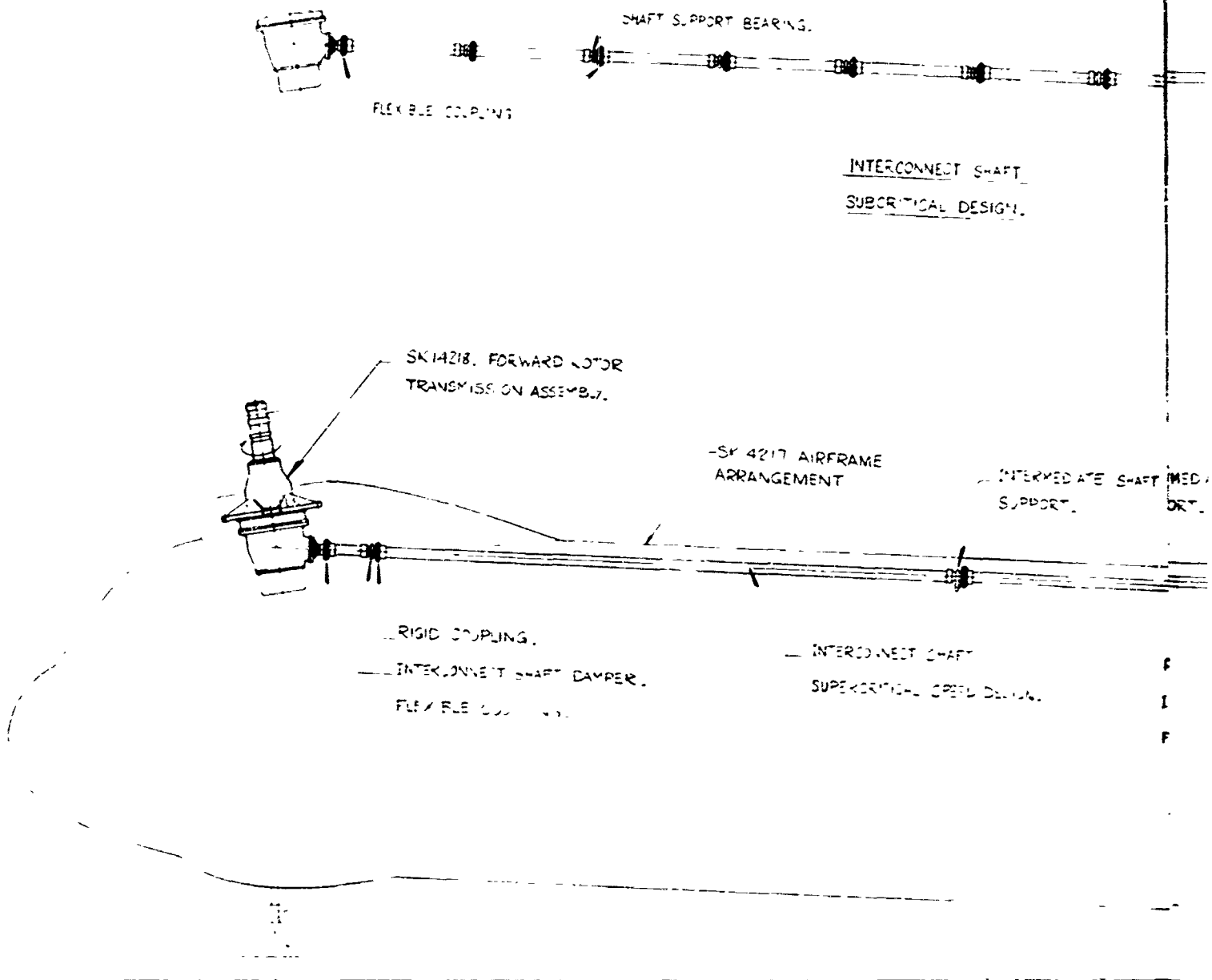
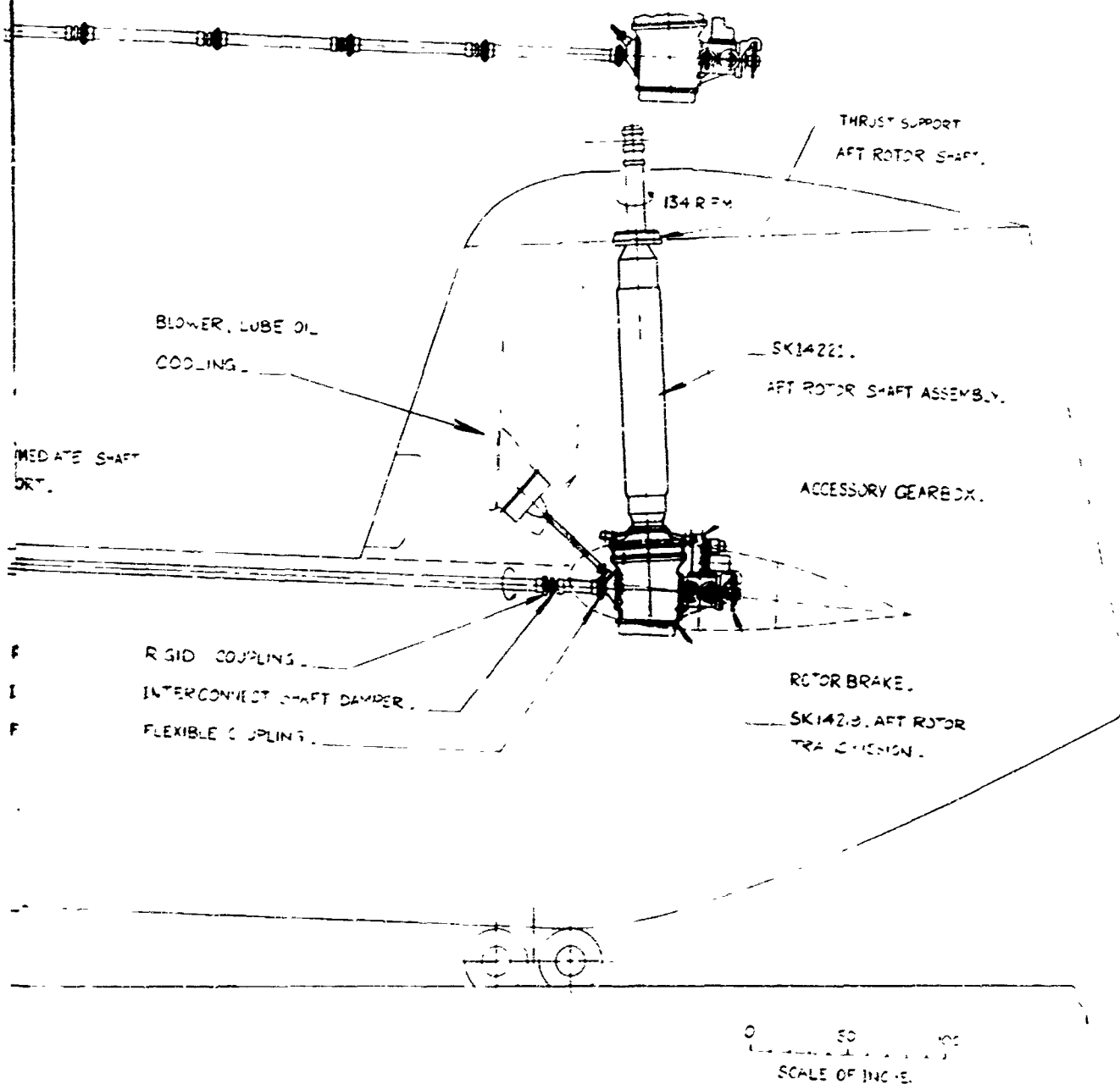


FIGURE 37. DRIVE SYSTEM - CONFIGURATION I - SK 14216.  
(Sheet 1 of 2)



**B**

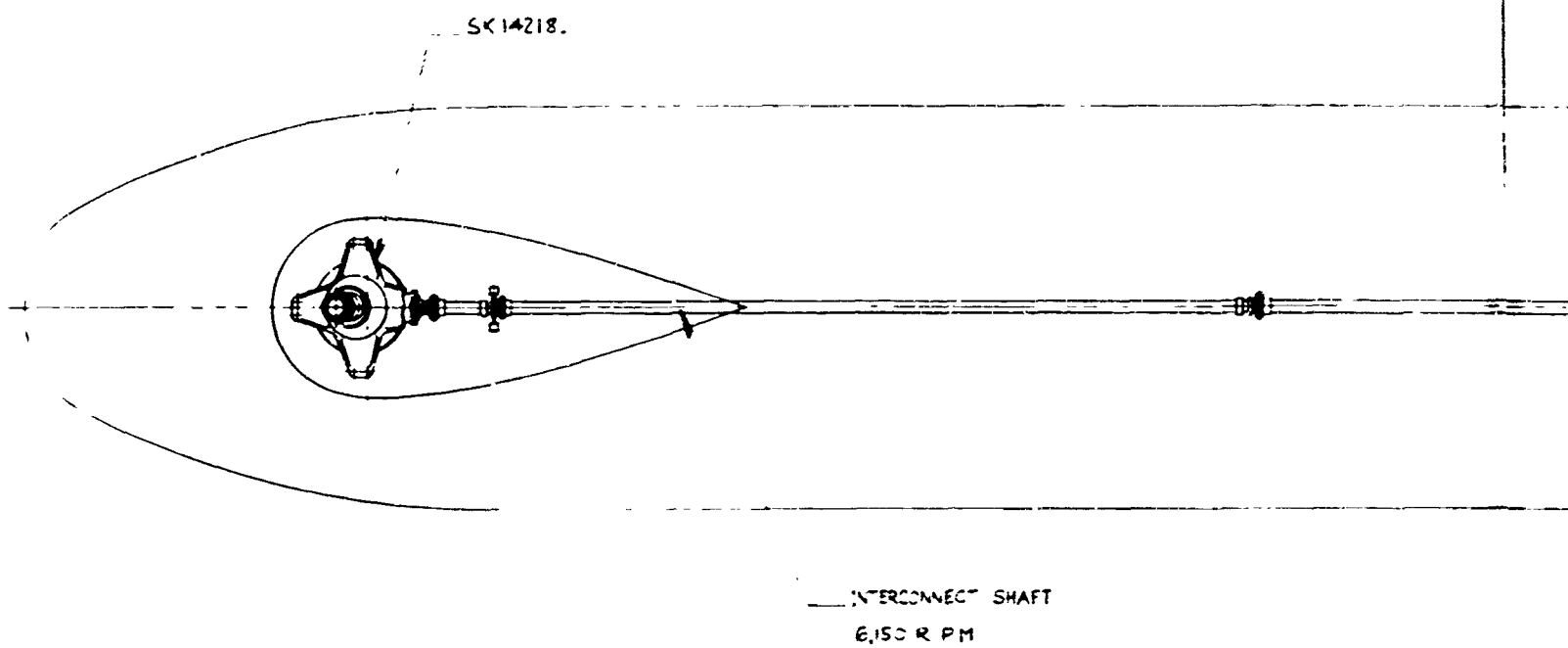


FIGURE 37 DRIVE SYSTEM - CONFIGURATION I - SK 14216.  
(Sheet 2 of 2)

ROTOR SHAFT THRUST SUPPORT.

SK. 14219.

SHAFT DAMPER.

ENGINE SHAFT 16,000 RPM

FLEXIBLE COUPLING.

POWER PLANT.

(T.55 SHOWN

T.6<sub>2</sub> ALTERNATE)

0 50 100  
SCALE OF INCHES.

**B**

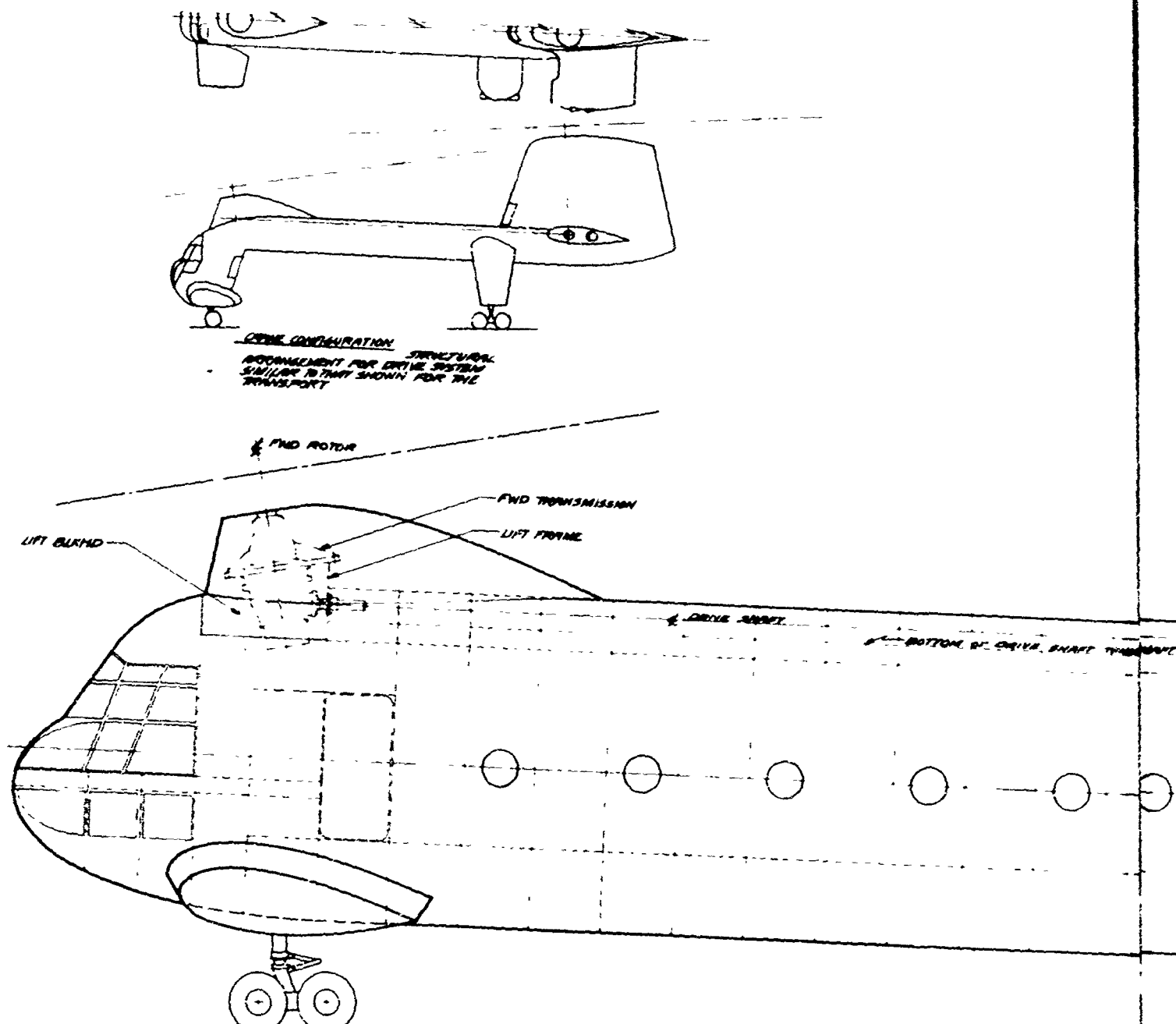
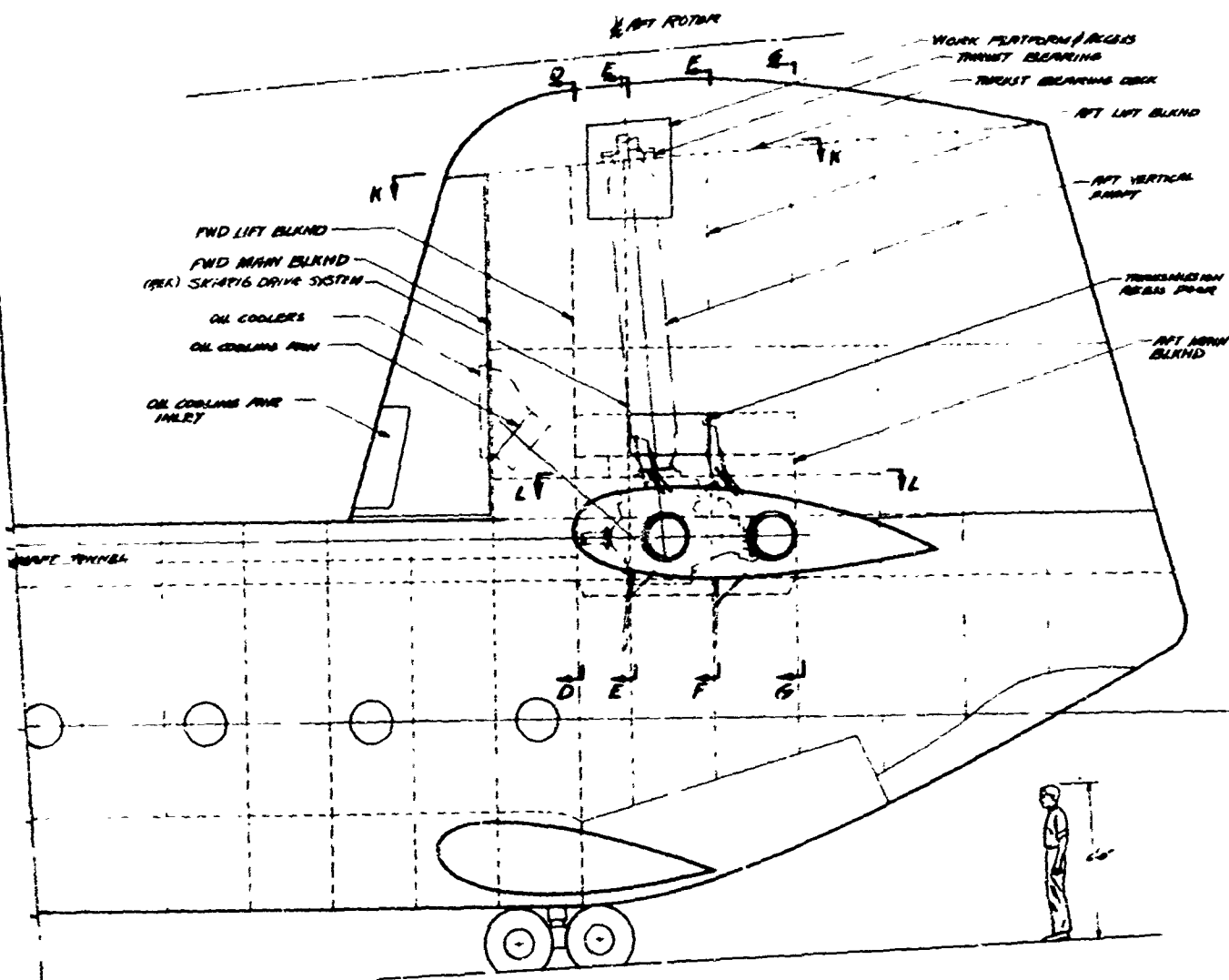


FIGURE 38. AIRFRAME ARRANGEMENT - CONFIGURATION I -  
 SK 14217. (Sheet 1 of 2)



**B**

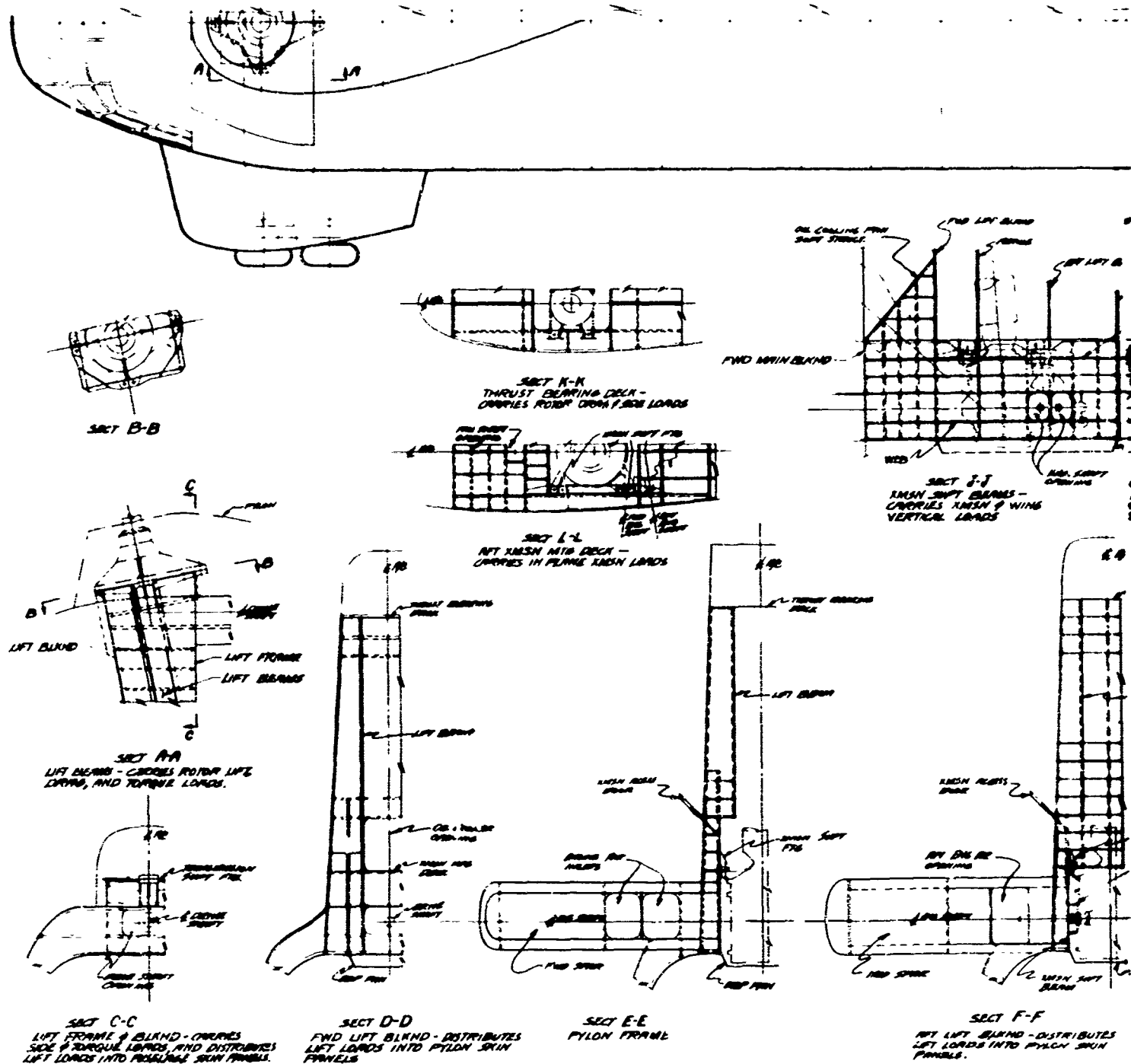


FIGURE 38. AIRFRAME ARRANGEMENT - CONFIGURATION I -  
SK 14217. (Sheet 2 of 2)



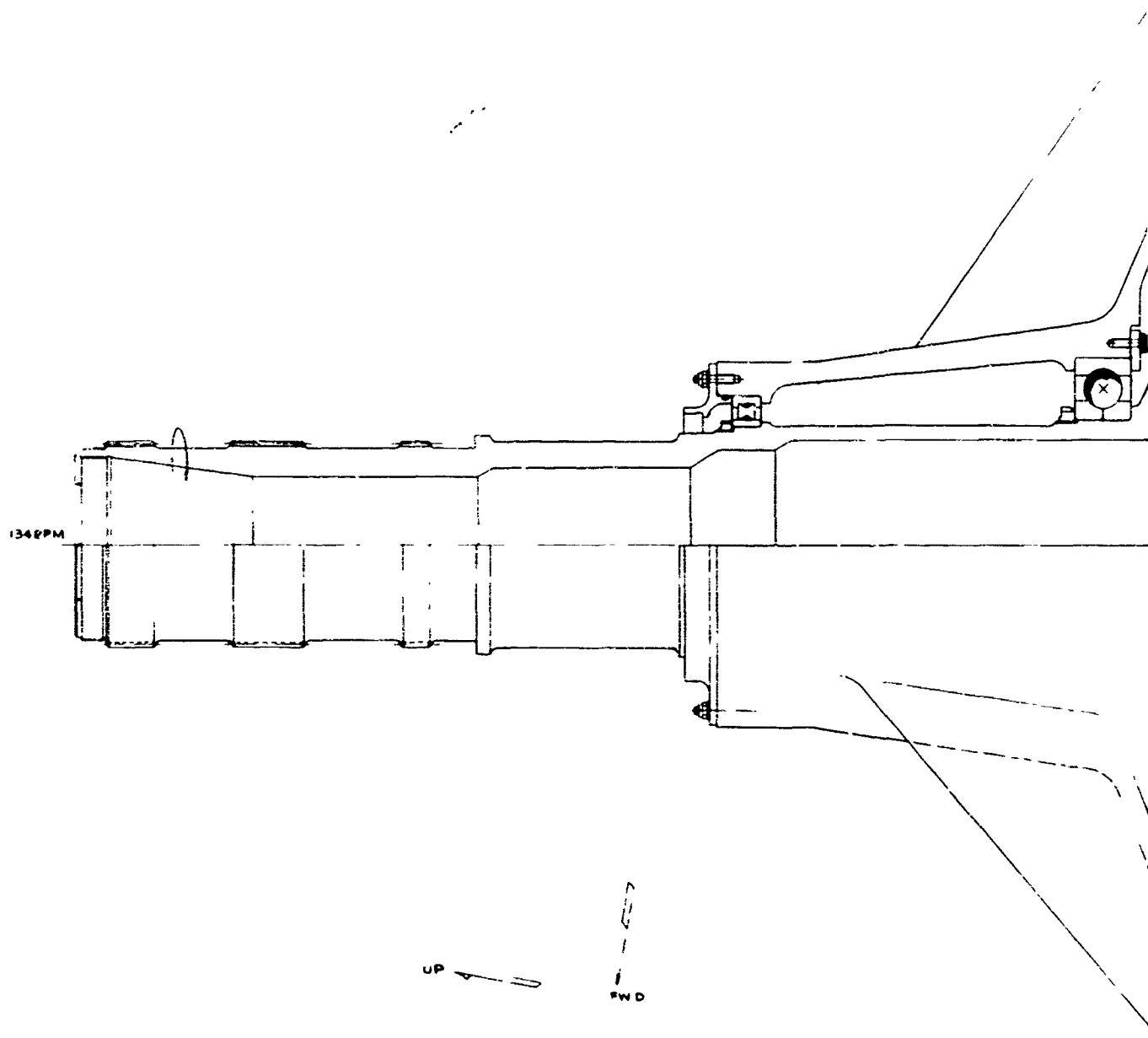
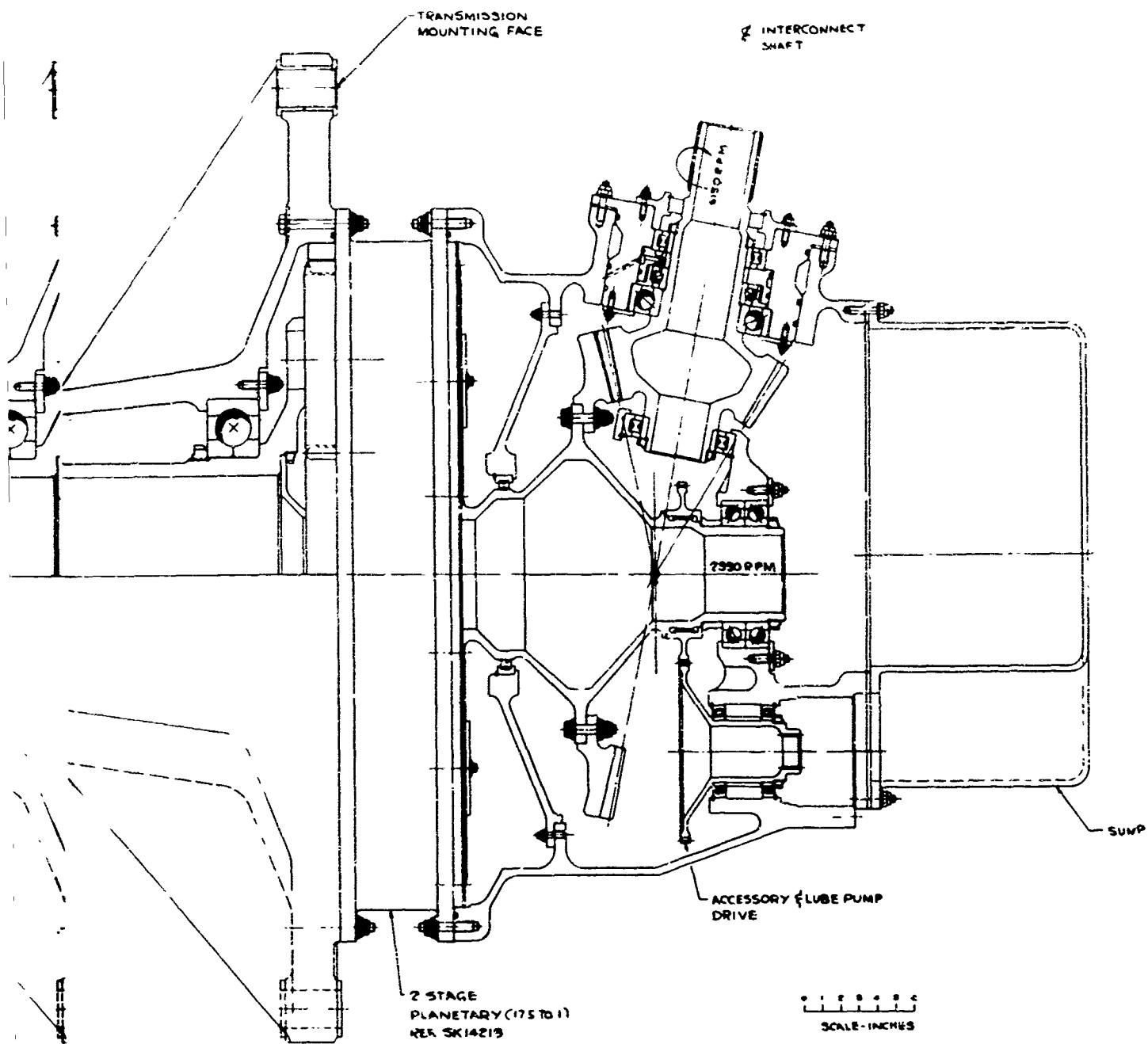


FIGURE 39. FORWARD ROTOR TRANSMISSION - CONFIGURATION I -  
SK 14218.



**B**

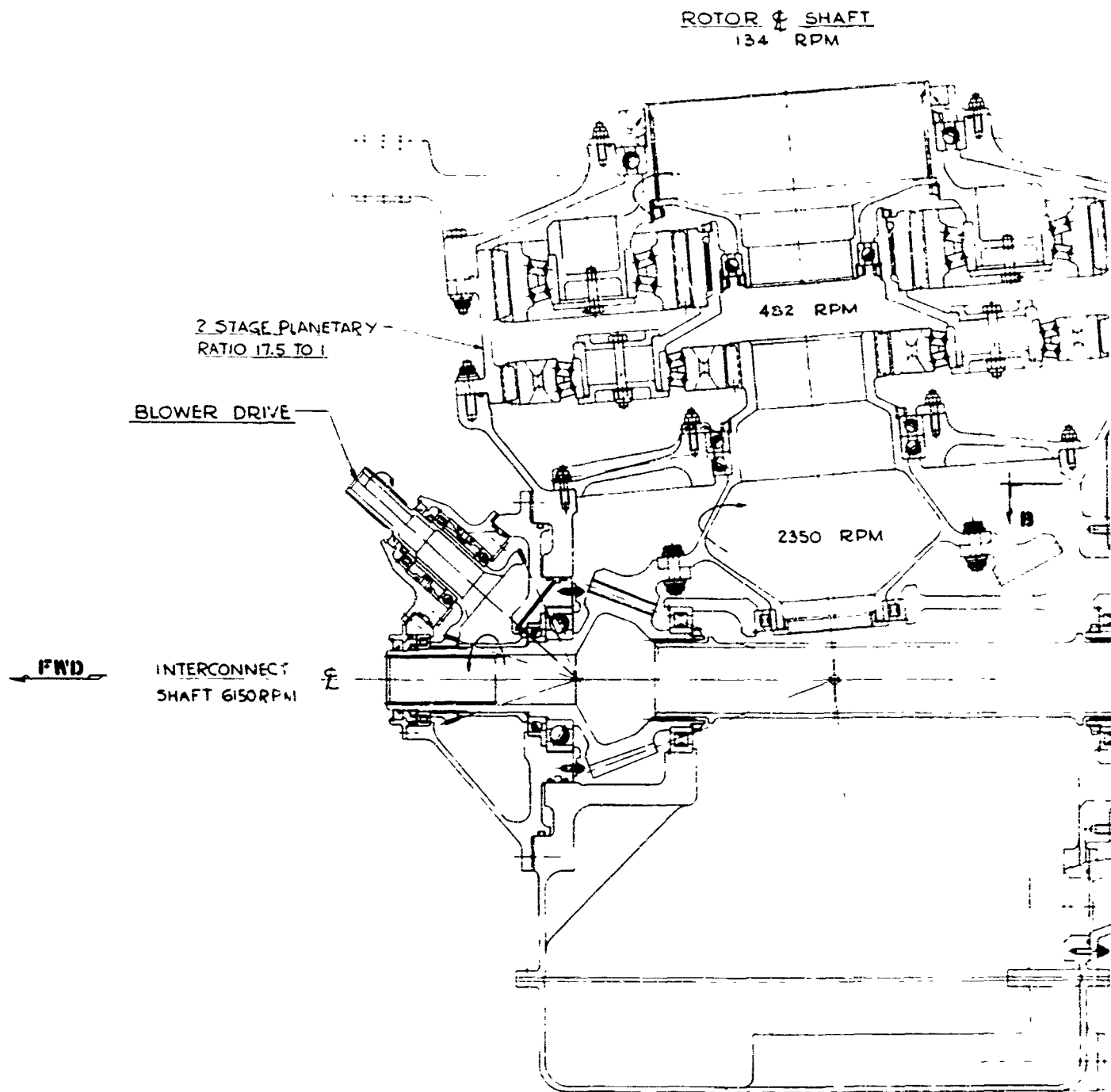
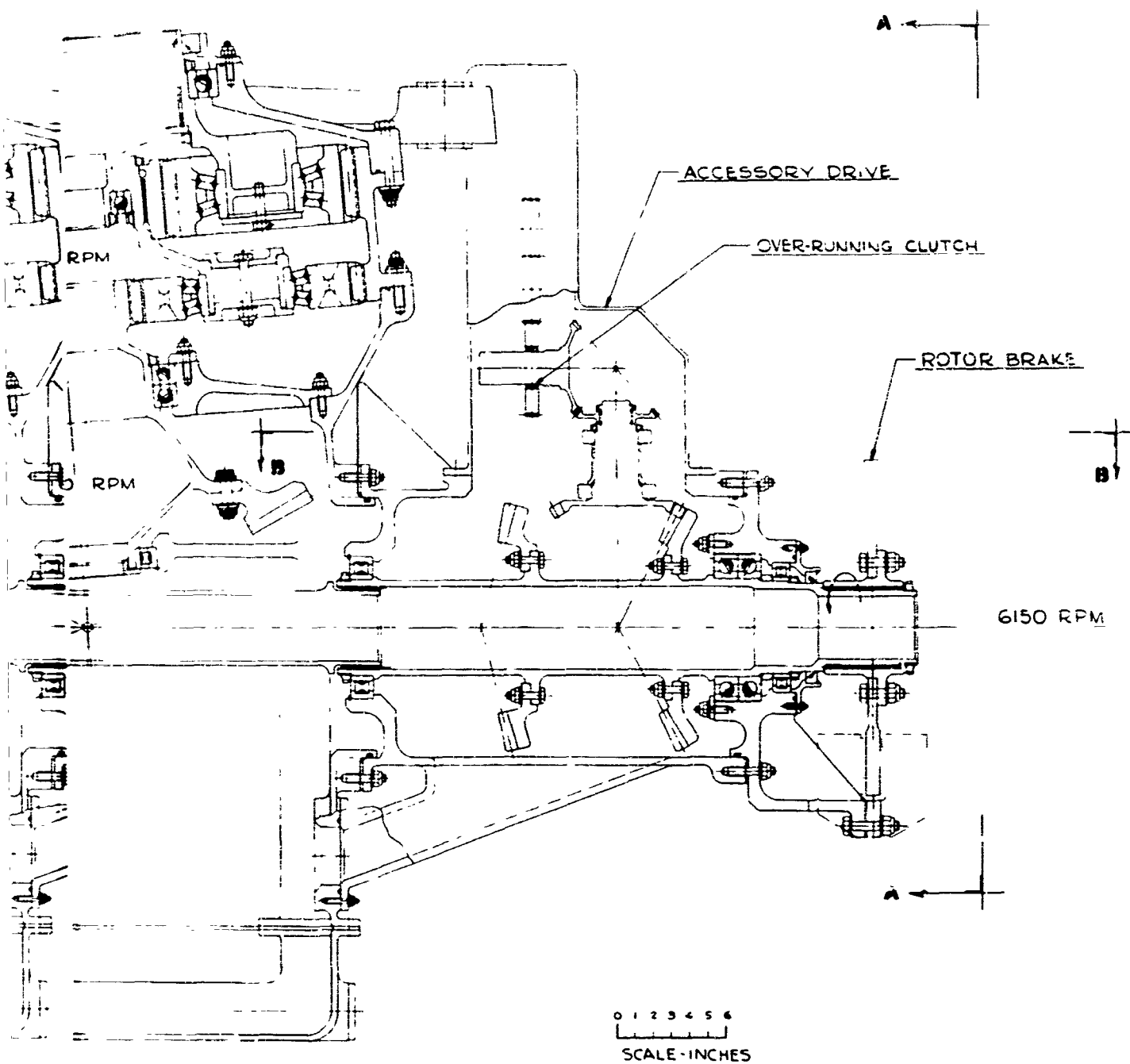


FIGURE 40. AFT ROTOR TRANSMISSION - CONFIGURATION I -  
SK 14219. (Sheet 1 of 3)

SHAFT  
PM



**B**

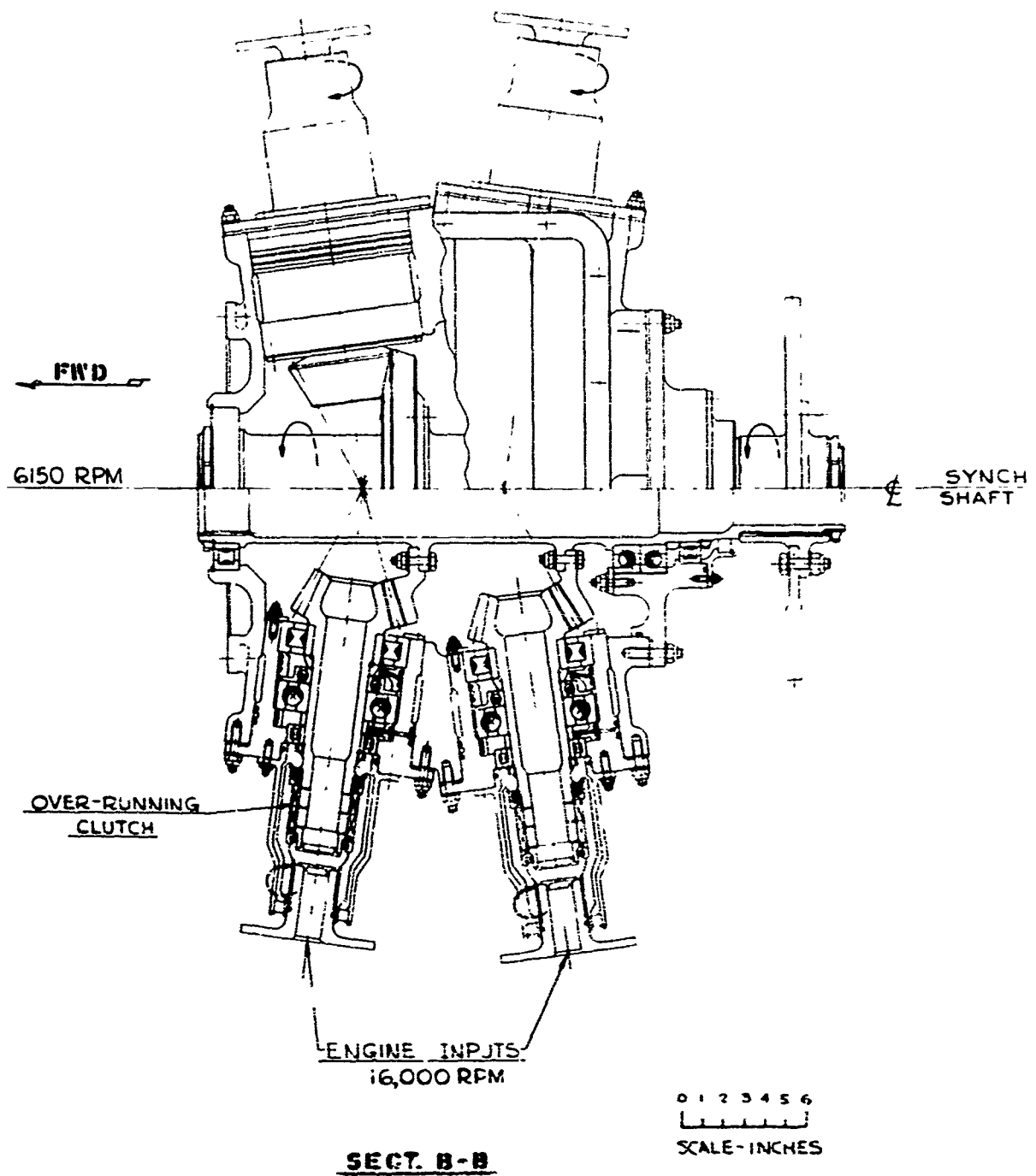


FIGURE 40. AFT ROTOR TRANSMISSION - CONFIGURATION I - SK 14219. (Sheet 2 of 3)

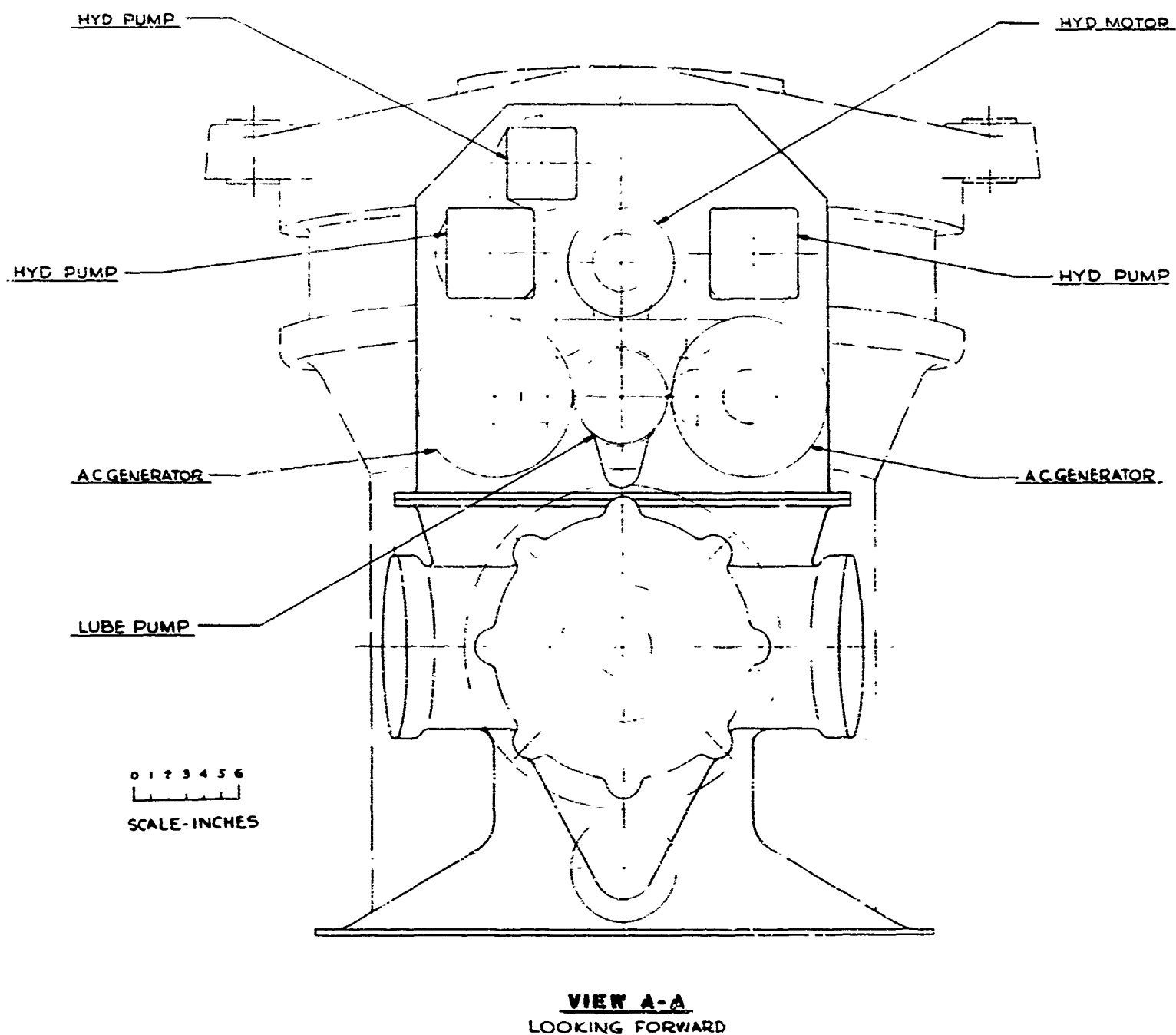


FIGURE 40. AFT ROTOR TRANSMISSION - CONFIGURATION I -  
SK 14219. (Sheet 3 of 3)

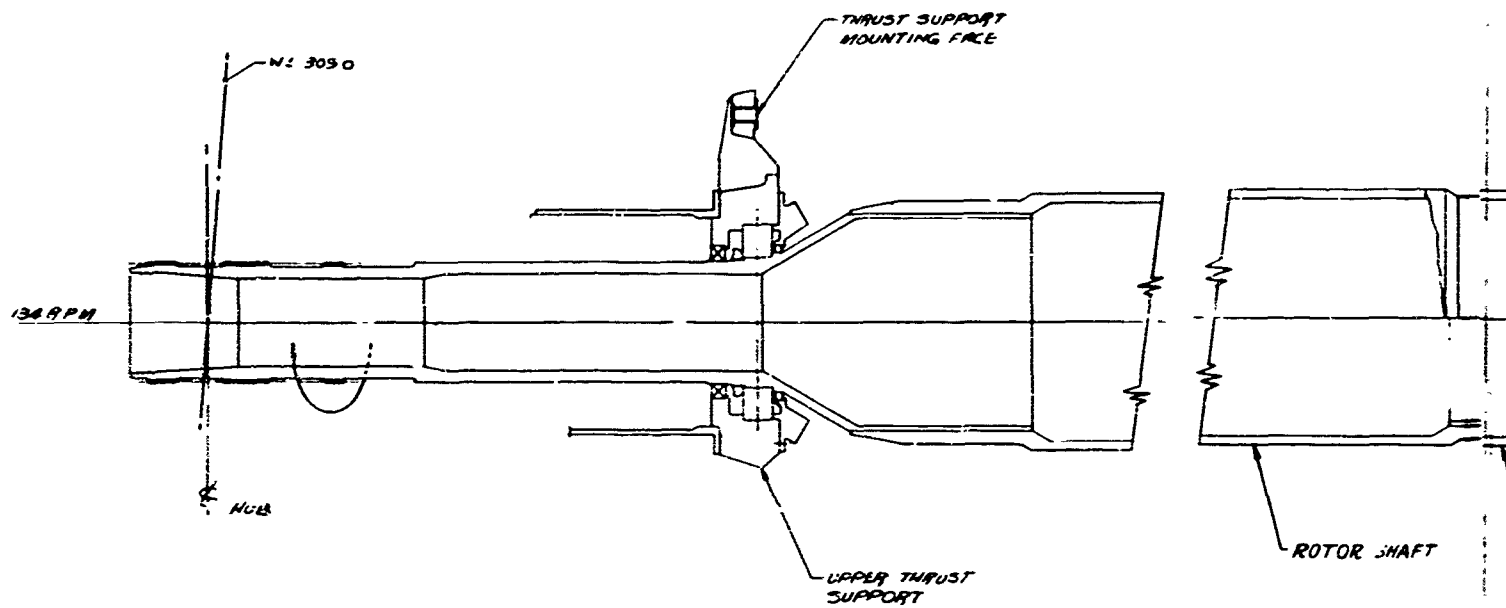
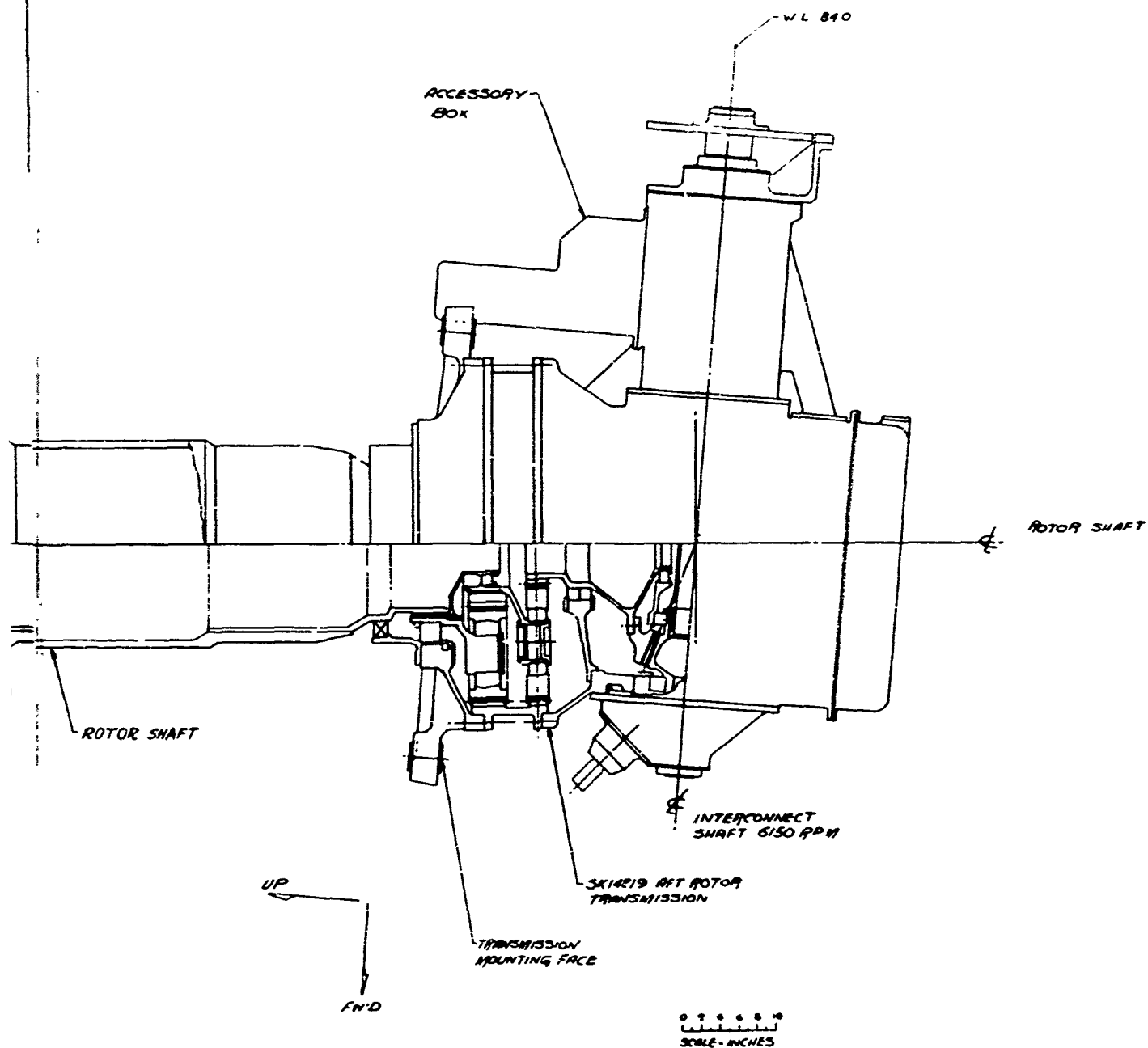


FIGURE 41. AFT ROTOR SHAFT - CONFIGURATION I - SK 14221.



**B**

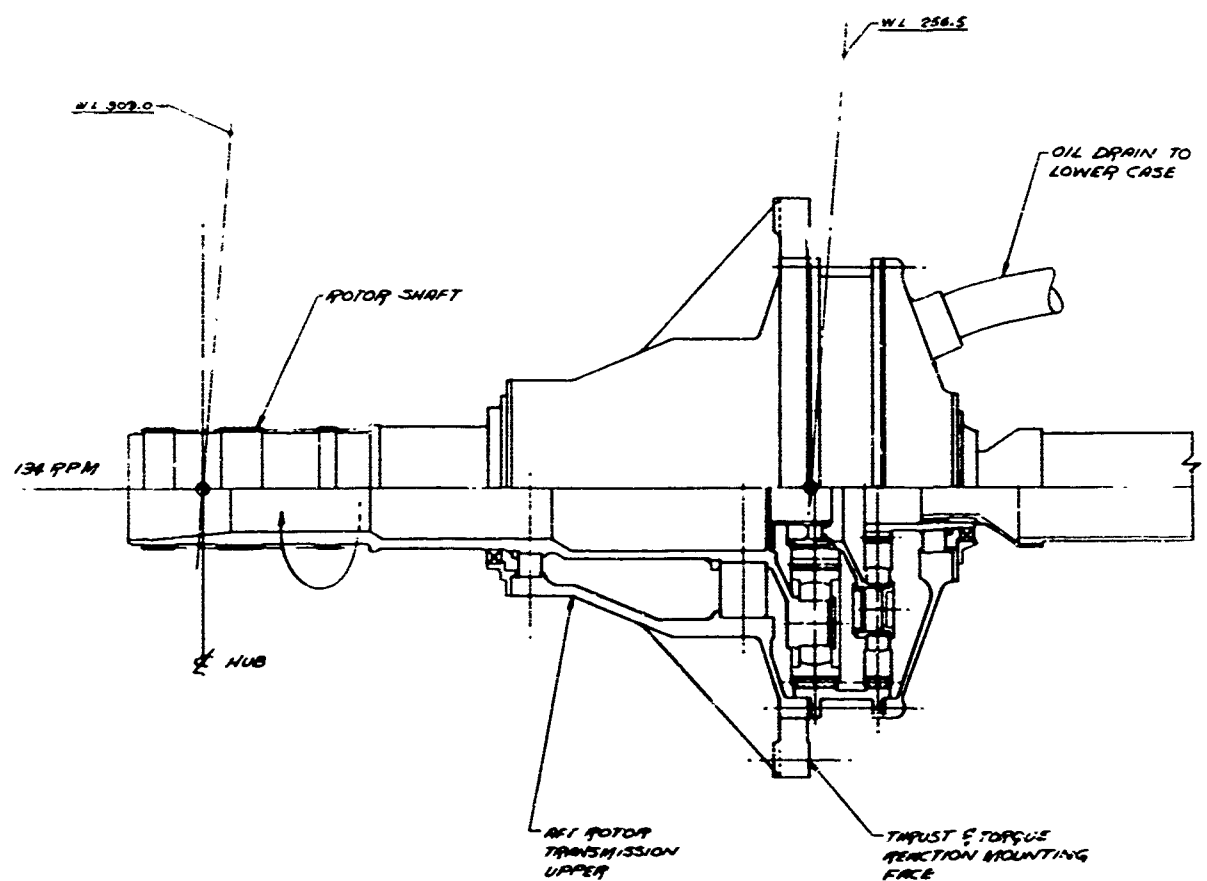
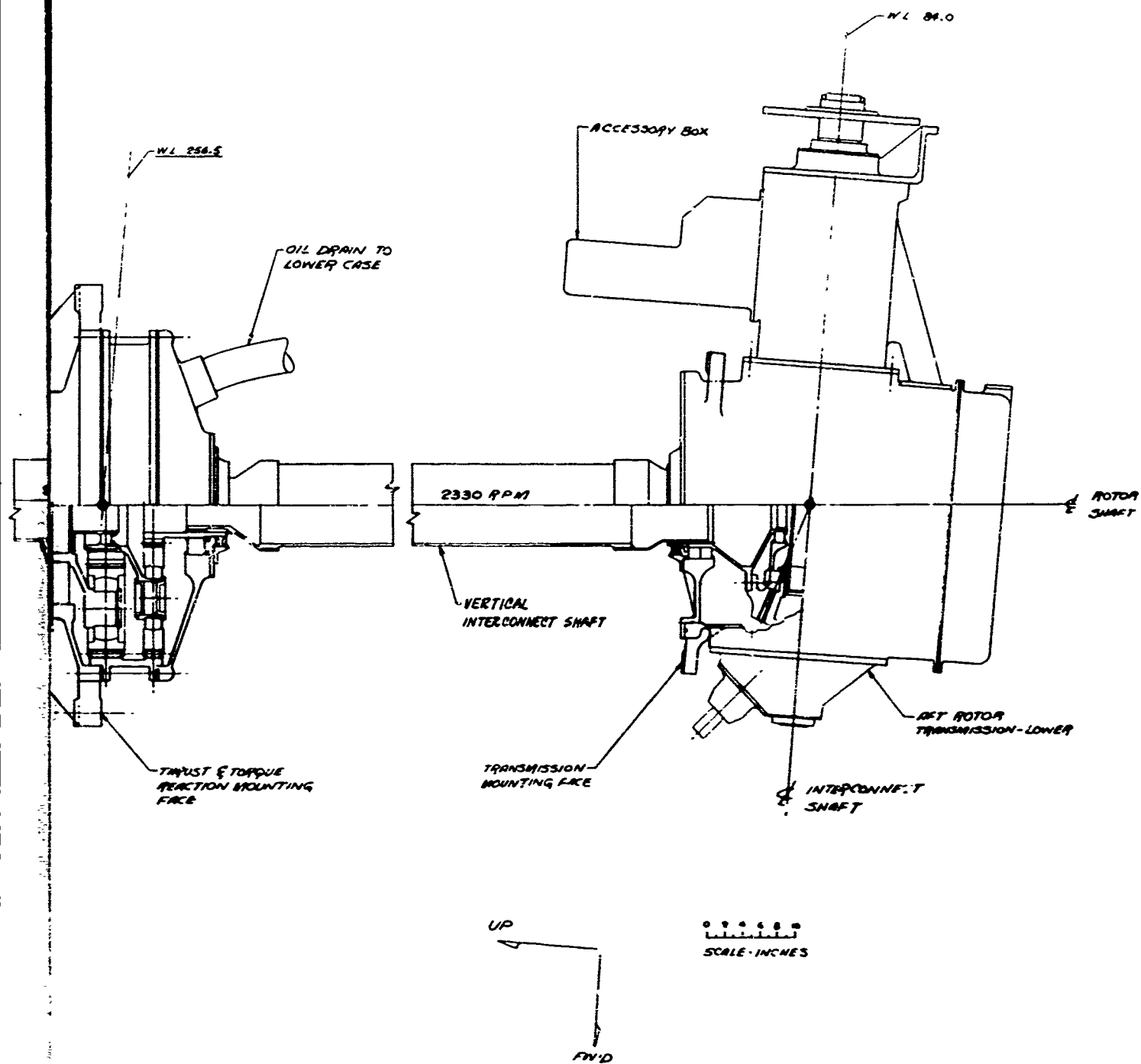


FIGURE 42. HIGH-MOUNTED REDUCTION - ALTERNATE CONFIGURATION  
IA - SK 14222.



ALTERNATE CONFIGURATION

**B**

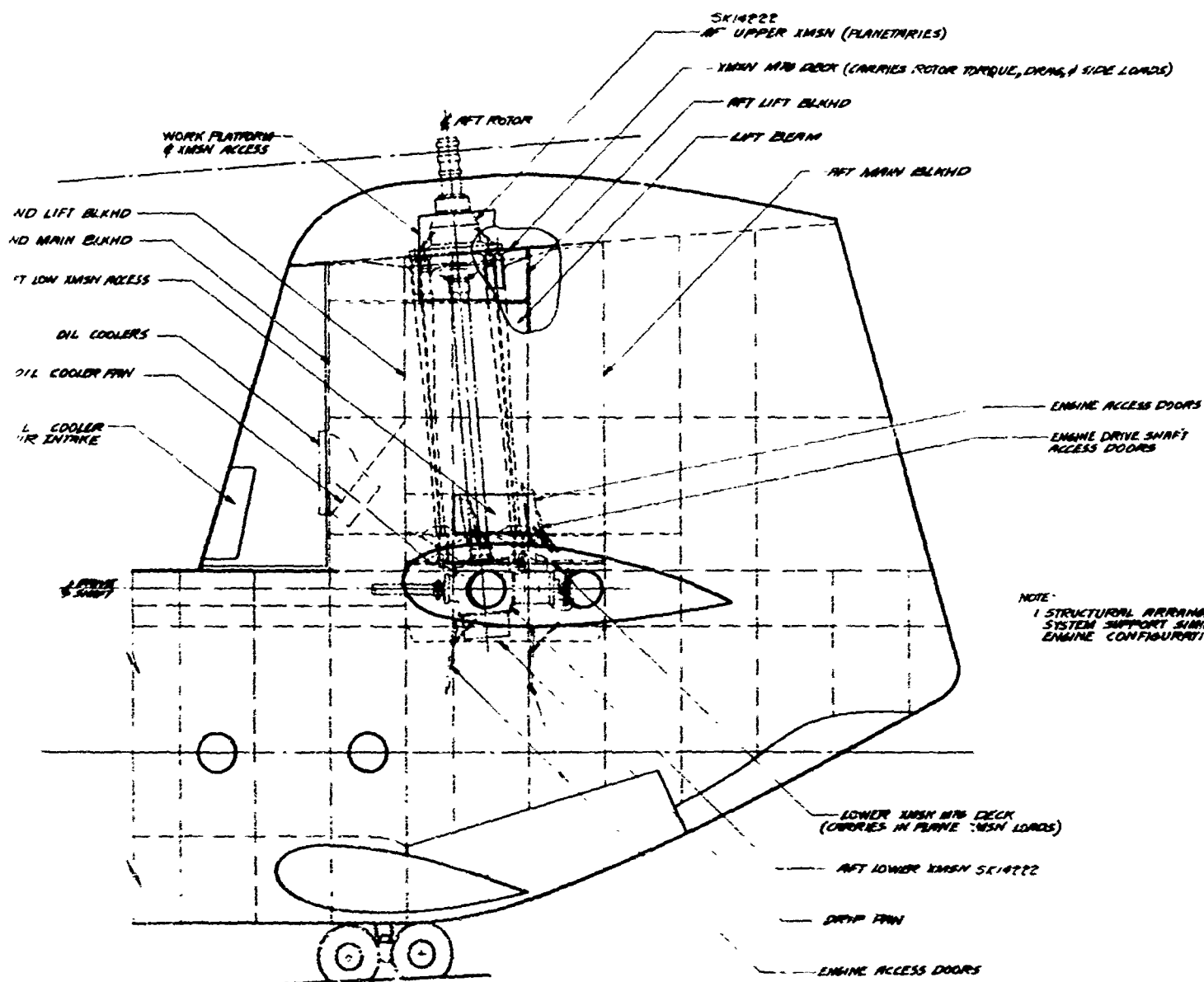


FIGURE 43. HIGH-MOUNTED REDUCTION AIRFRAME ARRANGEMENT -  
ALTERNATE CONFIGURATION 1A - SK 14225.

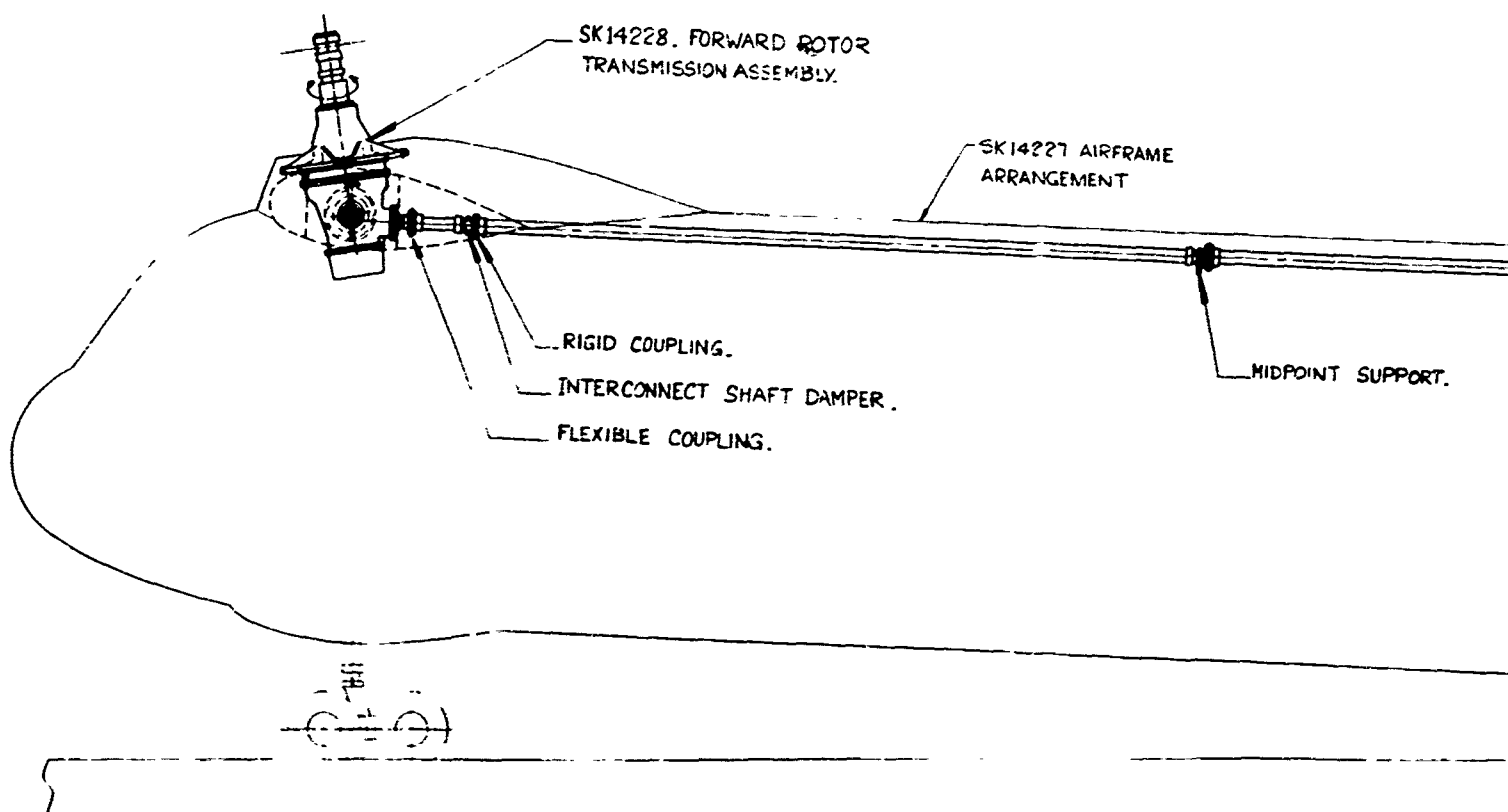
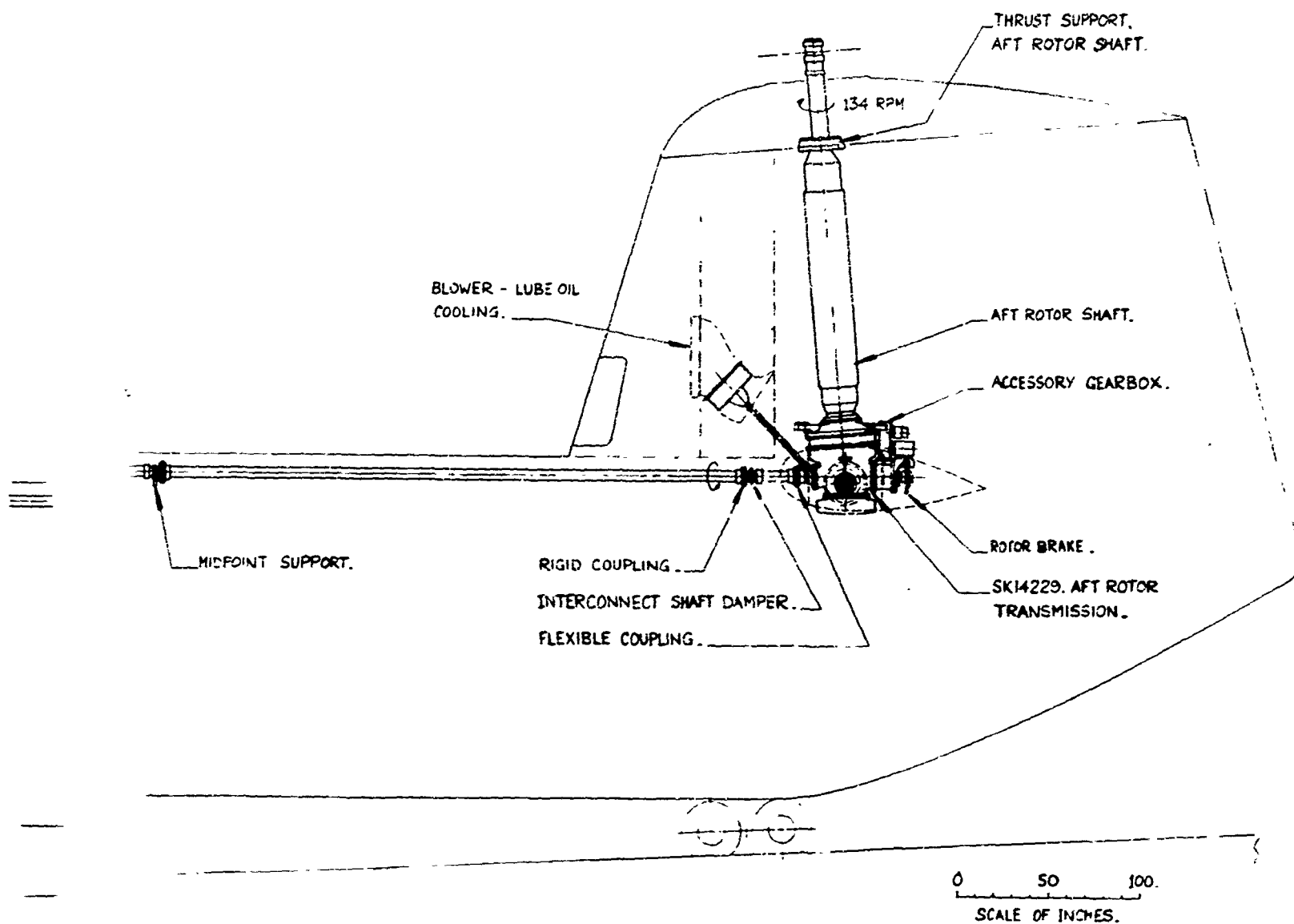


FIGURE 44. DRIVE SYSTEM - CONFIGURATION II - SK 14226.  
(Sheet 1 of 2)



**B**

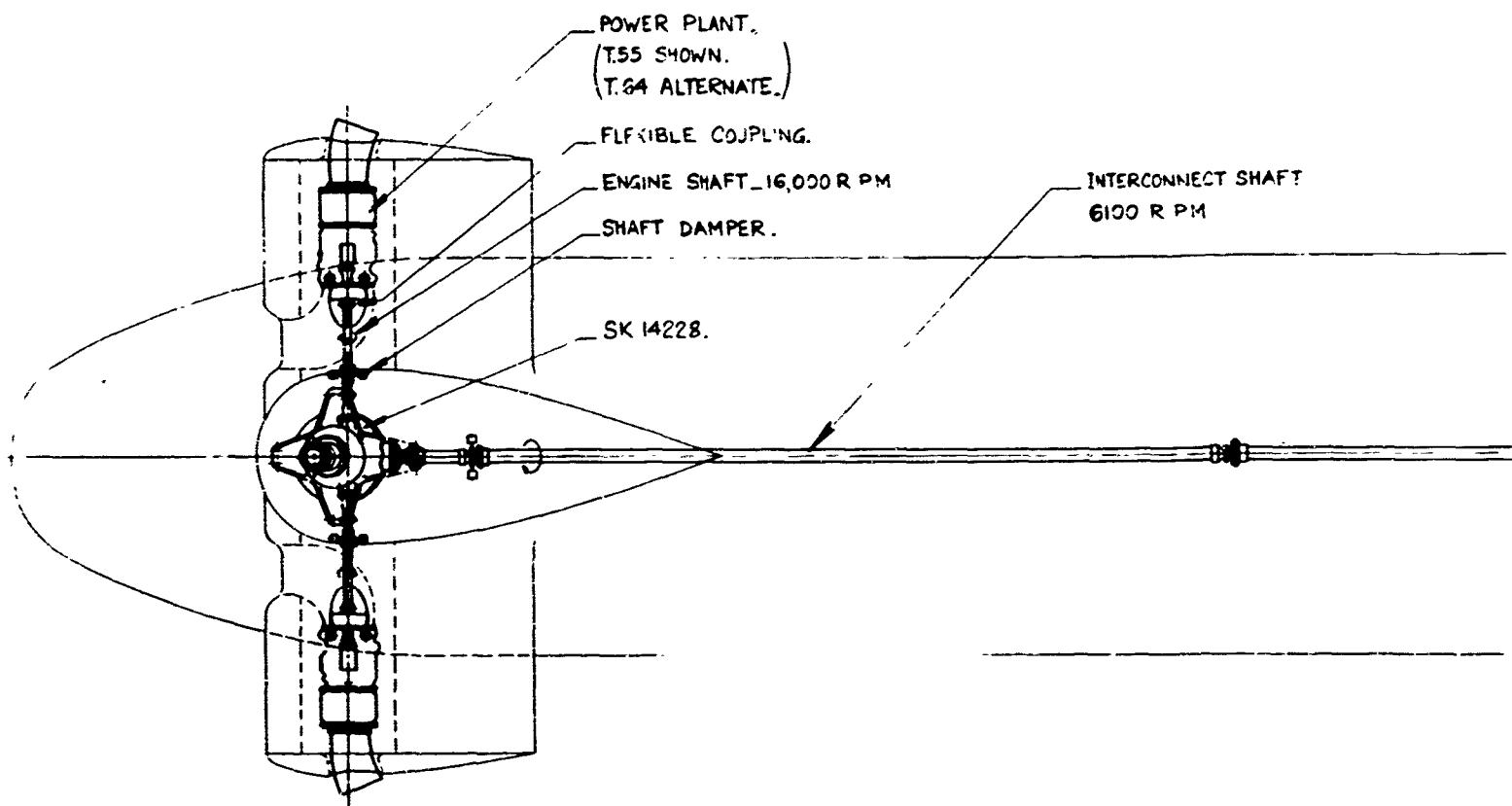
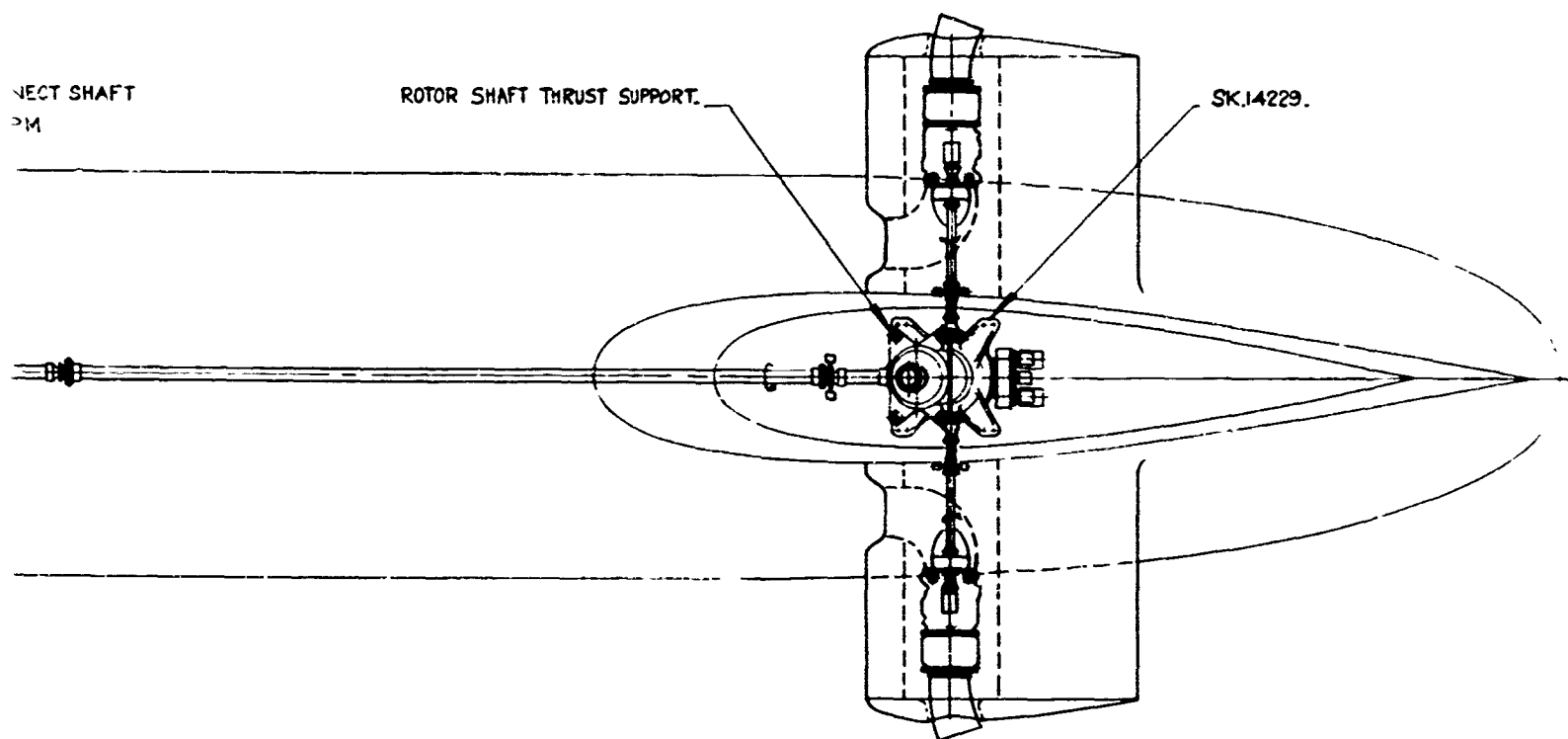


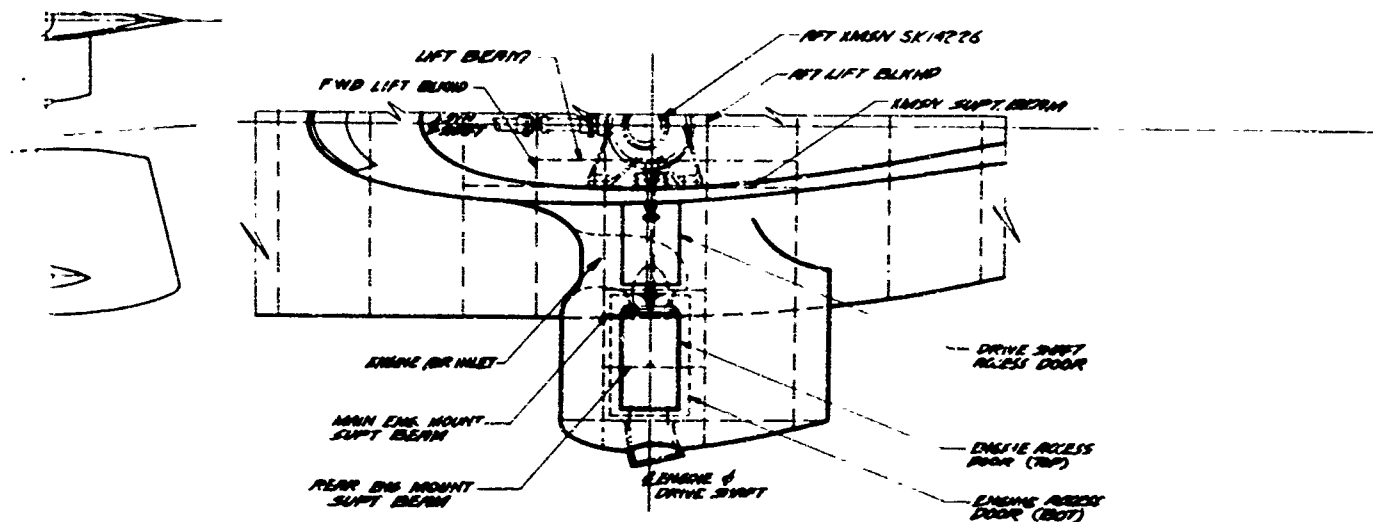
FIGURE 44. DRIVE SYSTEM - CONFIGURATION II - SK 14226.  
(Sheet 2 of 2)



0 50 100.  
SCALE OF INCHES.

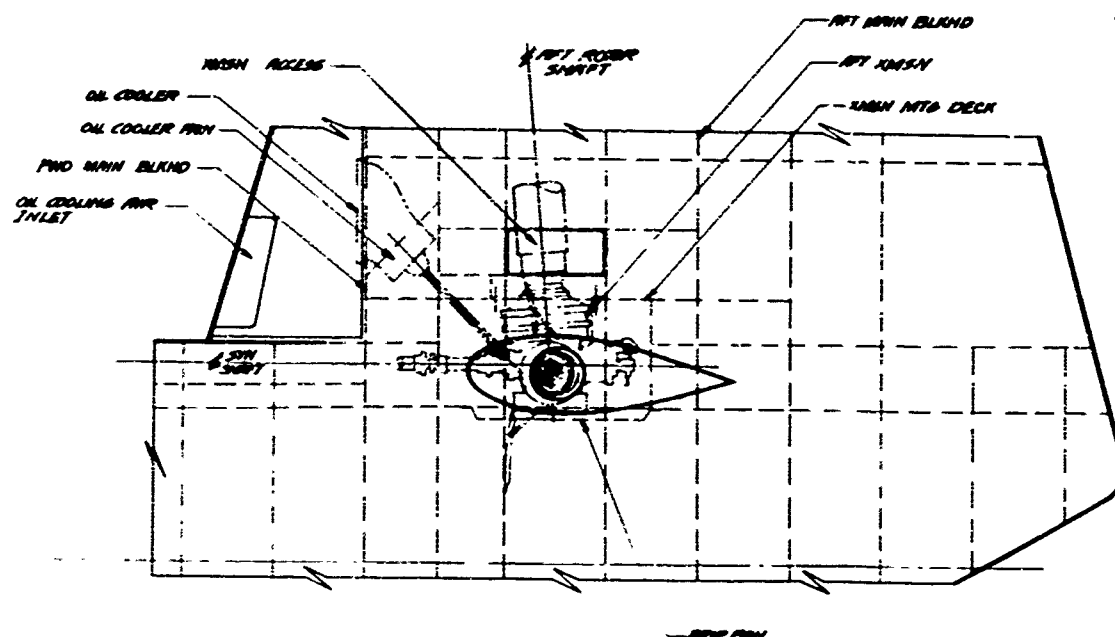
**B**





NOTE:

1. STRUCTURAL ARRANGEMENT FOR DRIVE SYSTEM SUPPORT SIMILAR TO THAT OF FOUR ENGINE COMPARTMENT
2. SECTIONS THRU RT PYLON & FUELLINE SIMILAR TO THOSE SHOWN ON SK-MBT



B

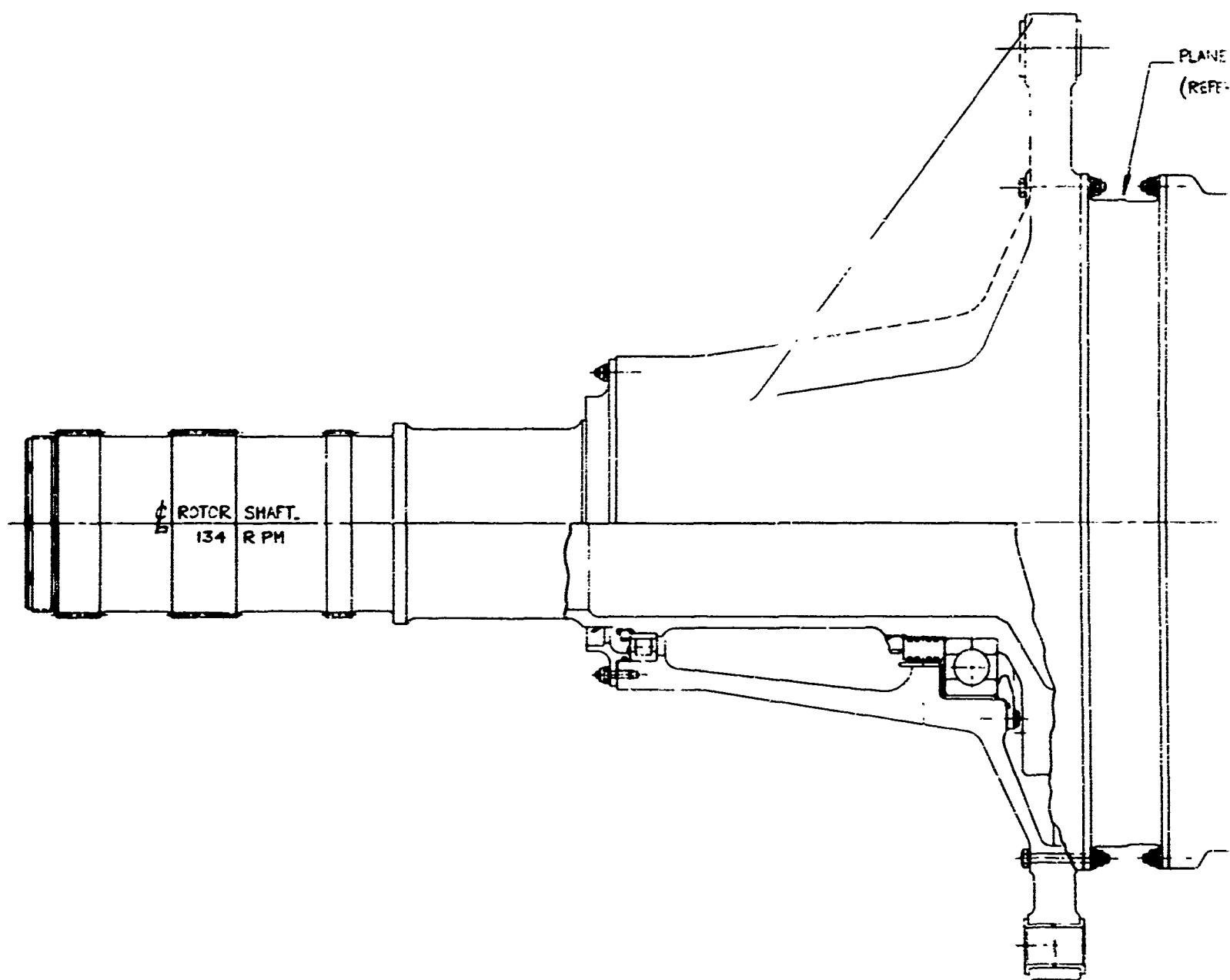
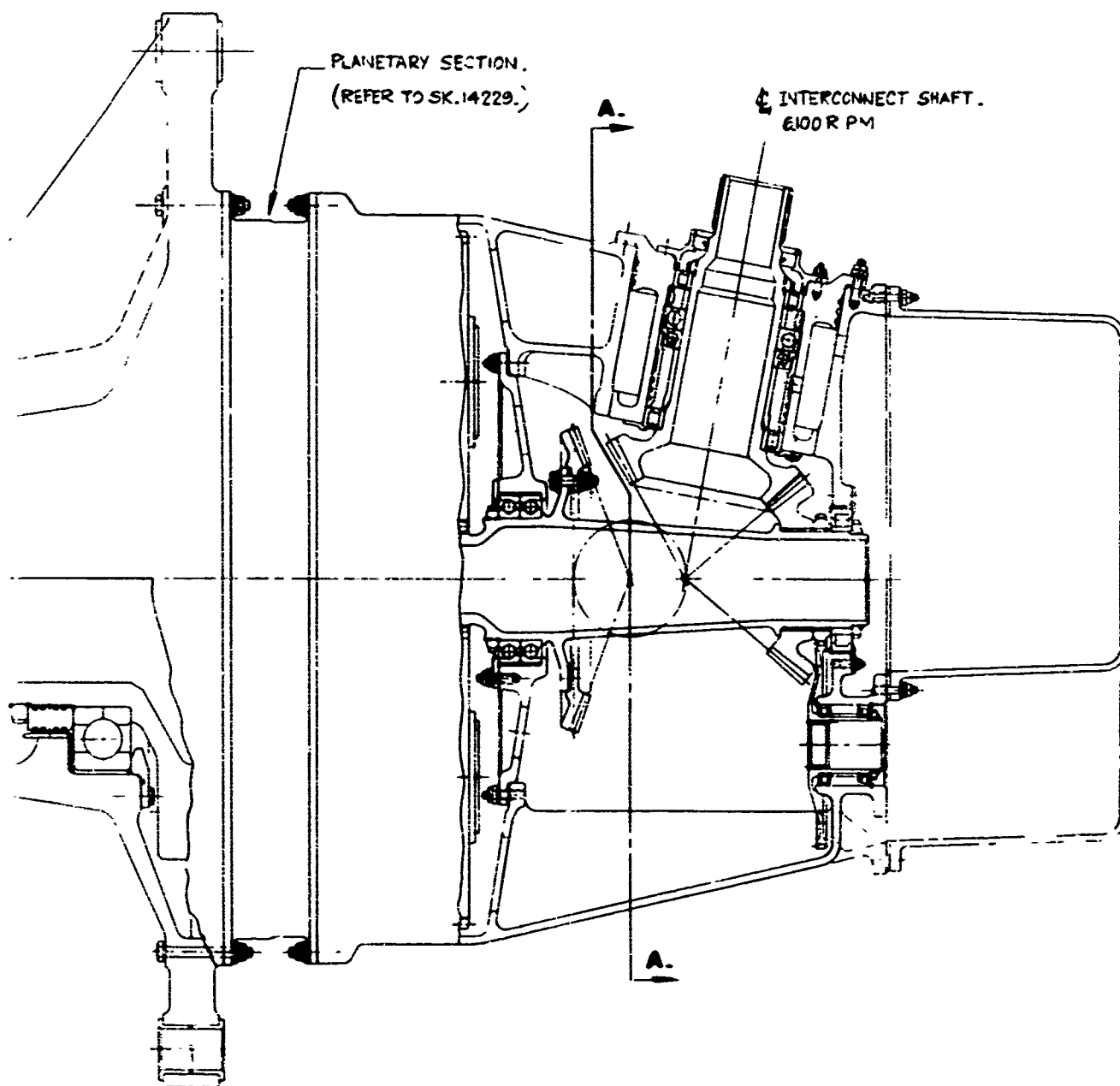


FIGURE 46. FORWARD ROTOR TRANSMISSION - CONFIGURATION  
 II - SK 14228. (Sheet 1 of 2)



**B**

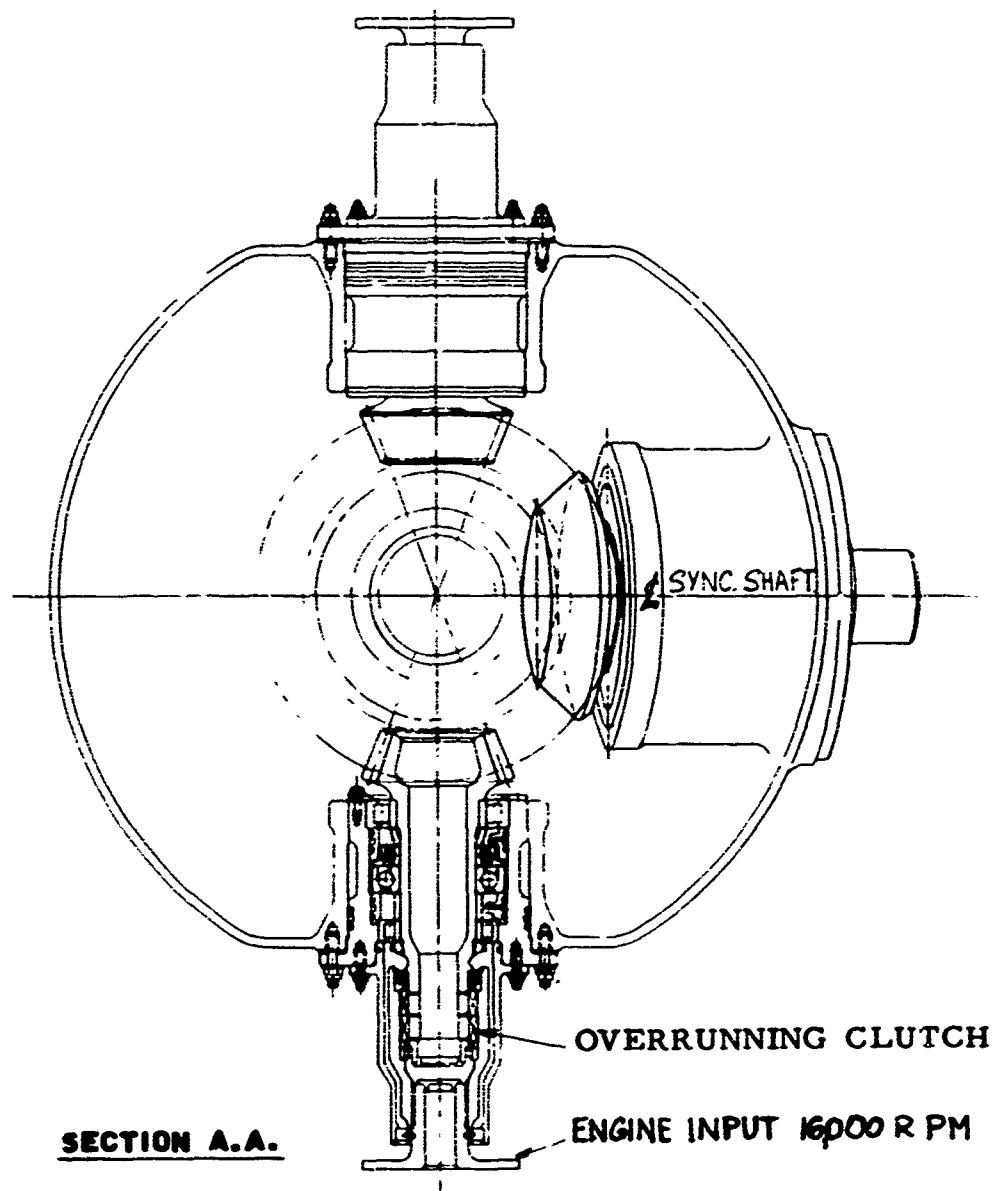


FIGURE 46. FORWARD ROTOR TRANSMISSION - CONFIGURATION II - SK 14228. (Sheet 2 of 2)

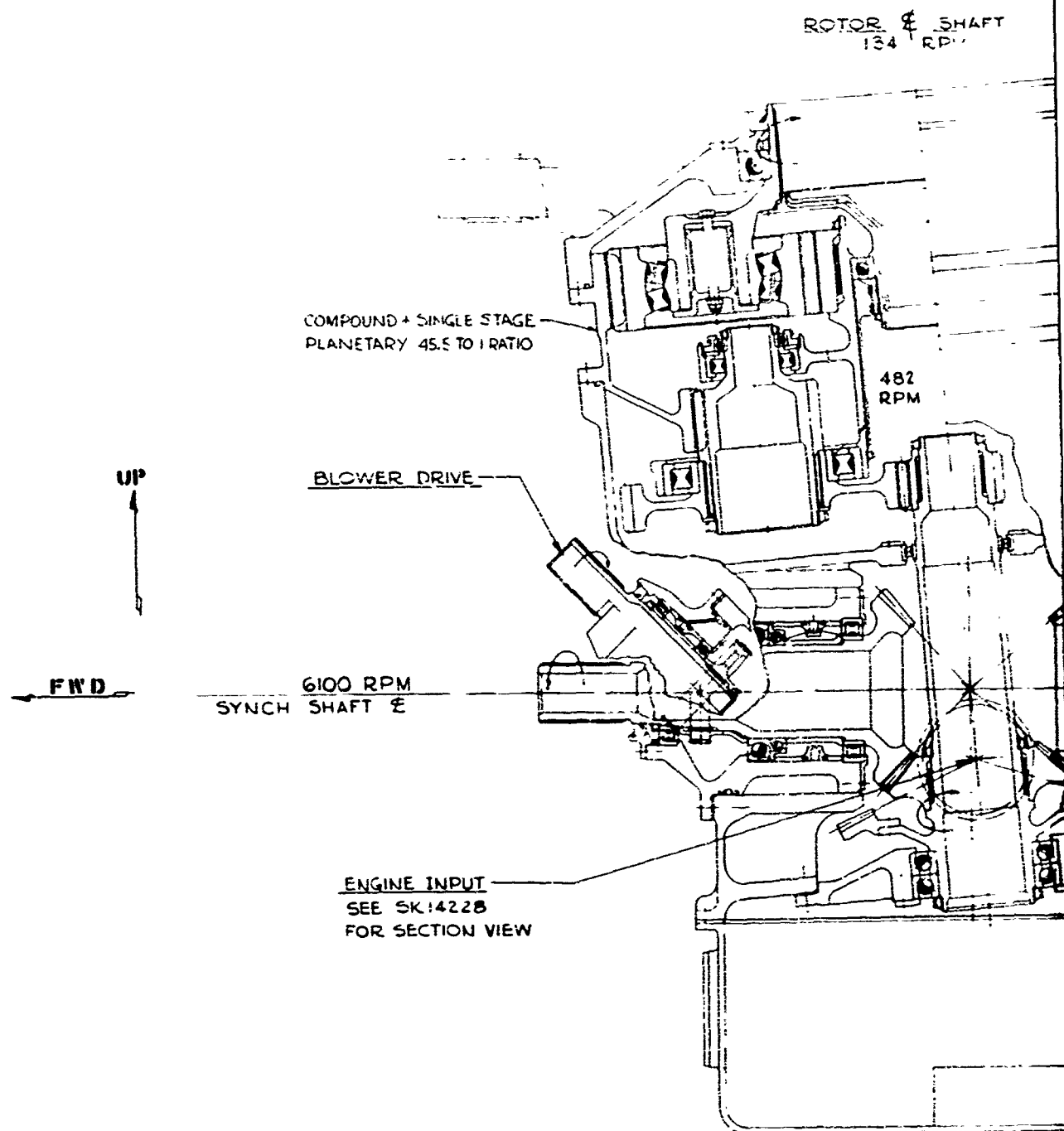


FIGURE 47. AFT ROTOR TRANSMISSION - CONFIGURATION II - SK 14229.

ROTOR & SHAFT  
134 RPM

482  
RPM

ACCESSORY DRIVE

ROTOR BRAKE

0 1 2 3 4 5 6  
SCALE-INCHES

FIGURATION II -

**B**

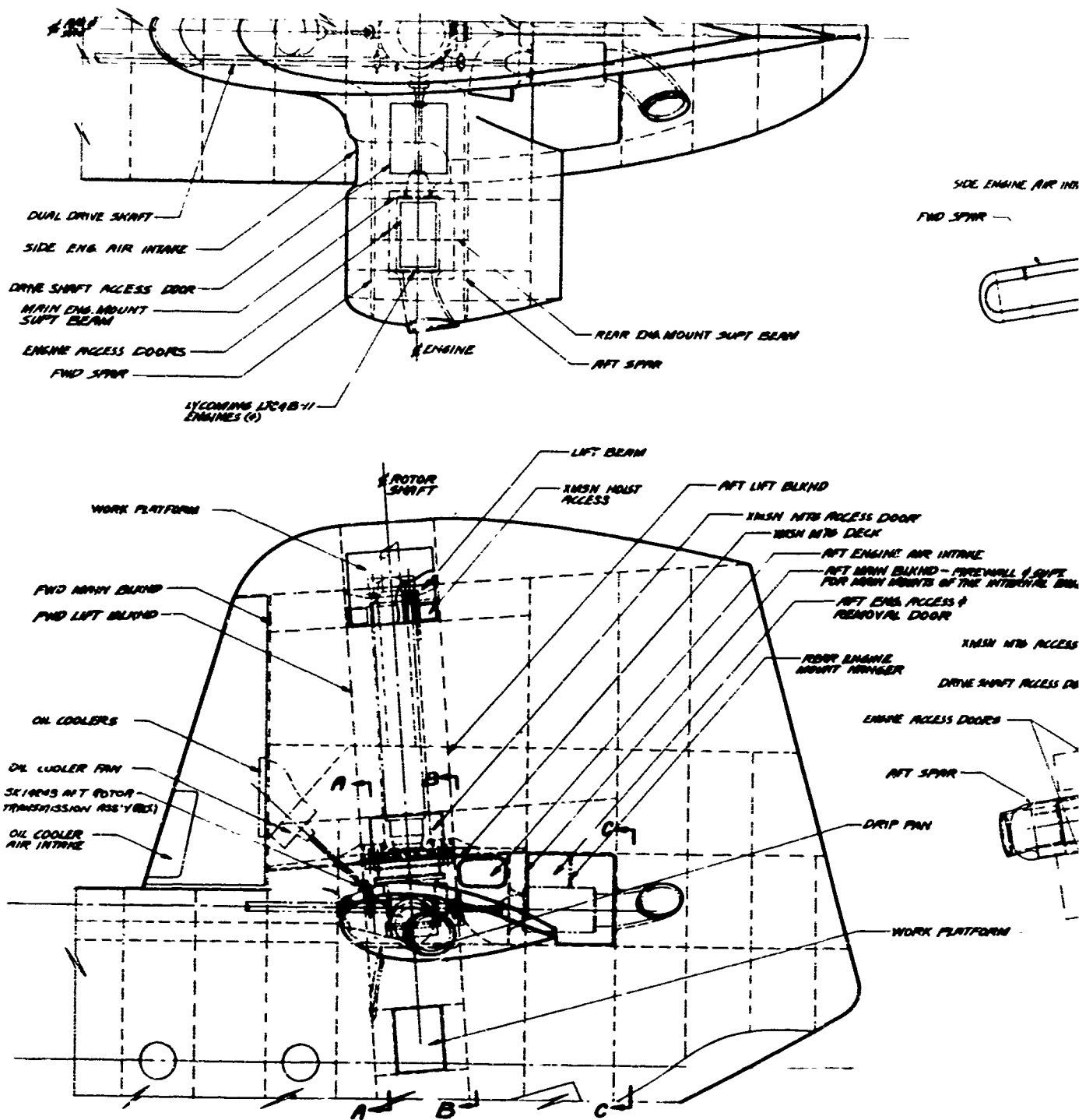
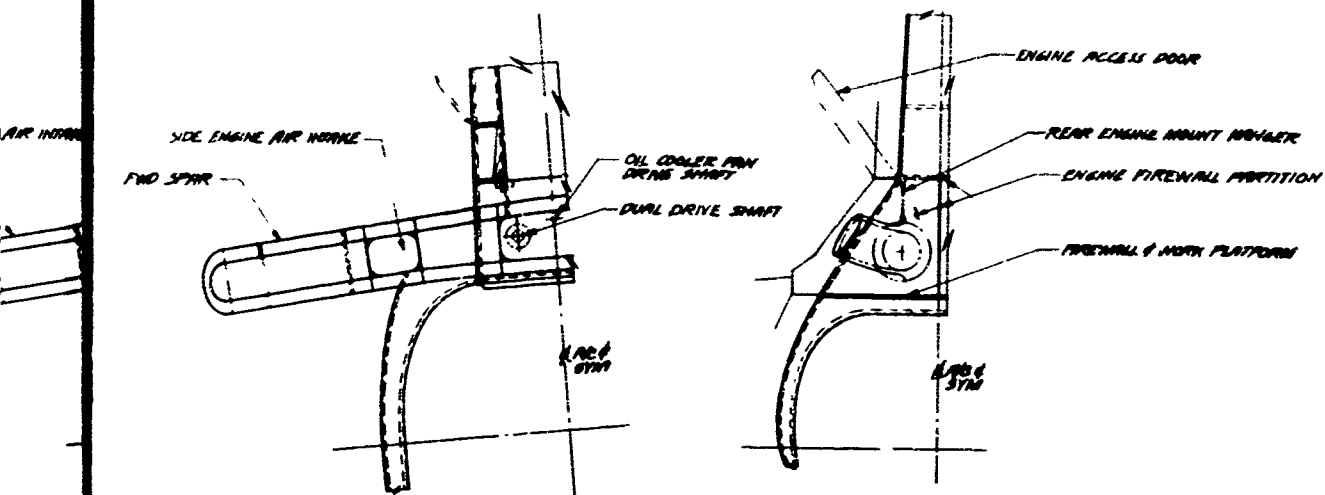
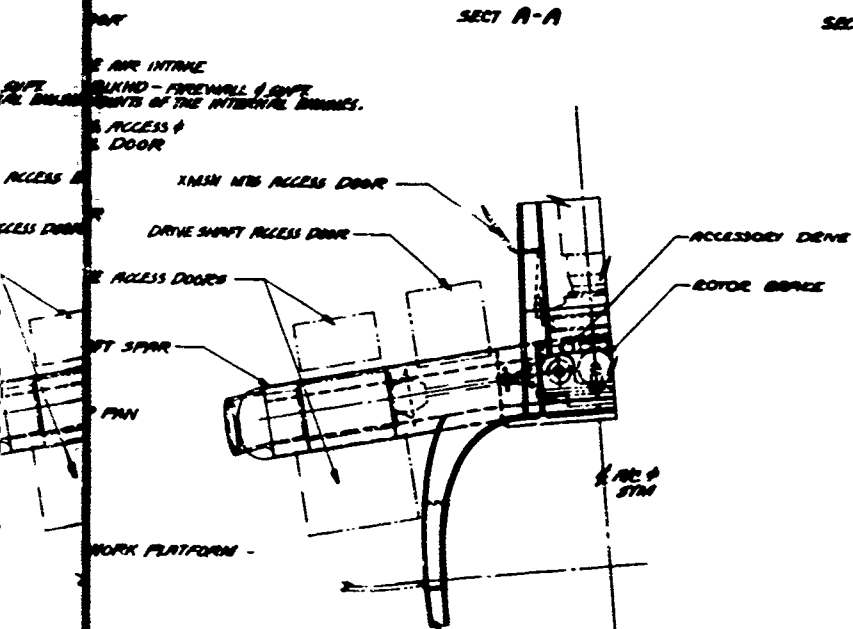


FIGURE 48. AIRFRAME ARRANGEMENT - CONFIGURATION III - SK 14242.



SECTION A-A

SECTION C-C



SECTION B-B  
(AFT LIFT BLIND)

NOTE:  
1. FWD FAN STRUCTURE SIMILAR TO THAT SHOWN FOR THE FOUR ENGINE ARRANGEMENT SR-16817.

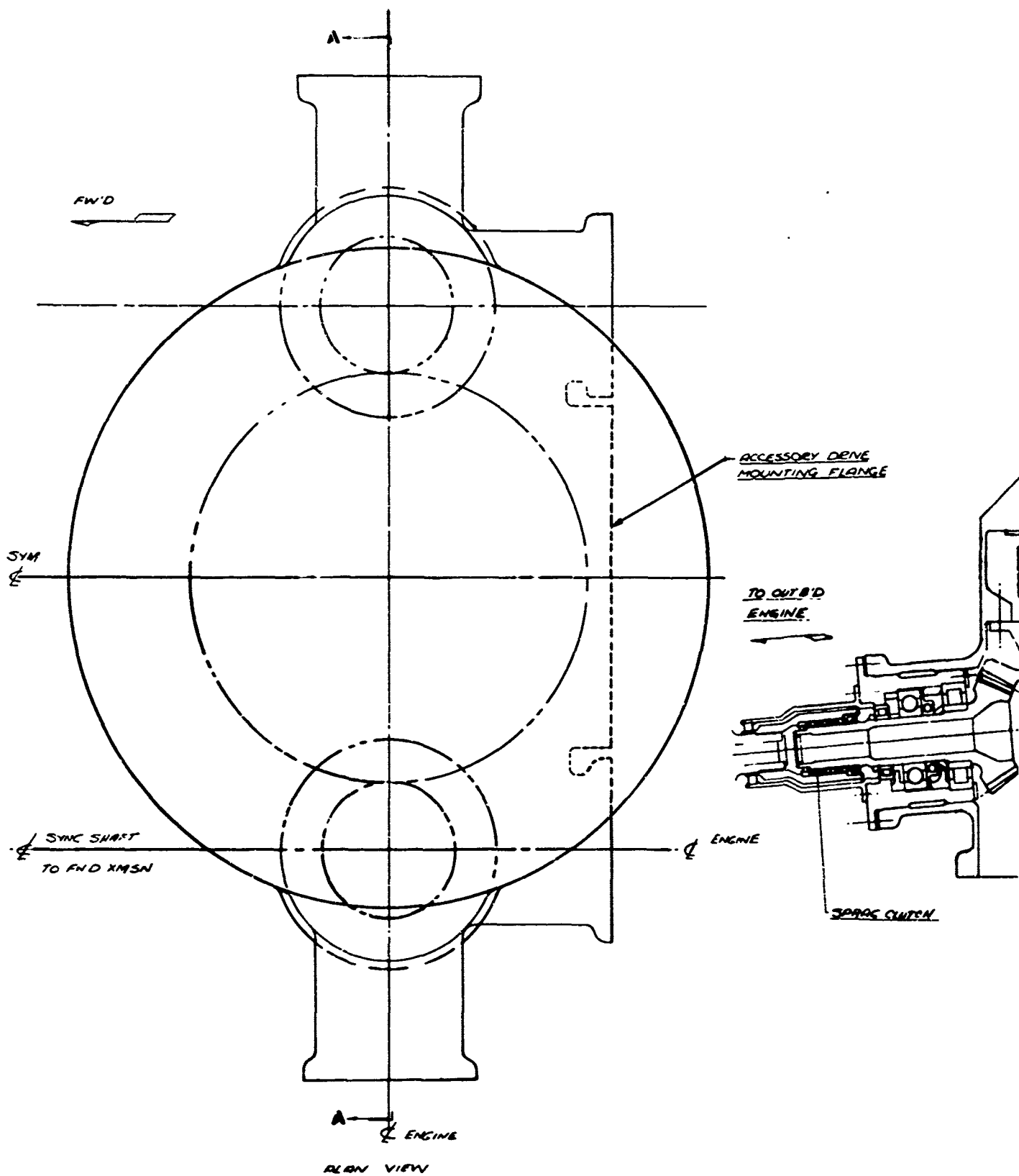
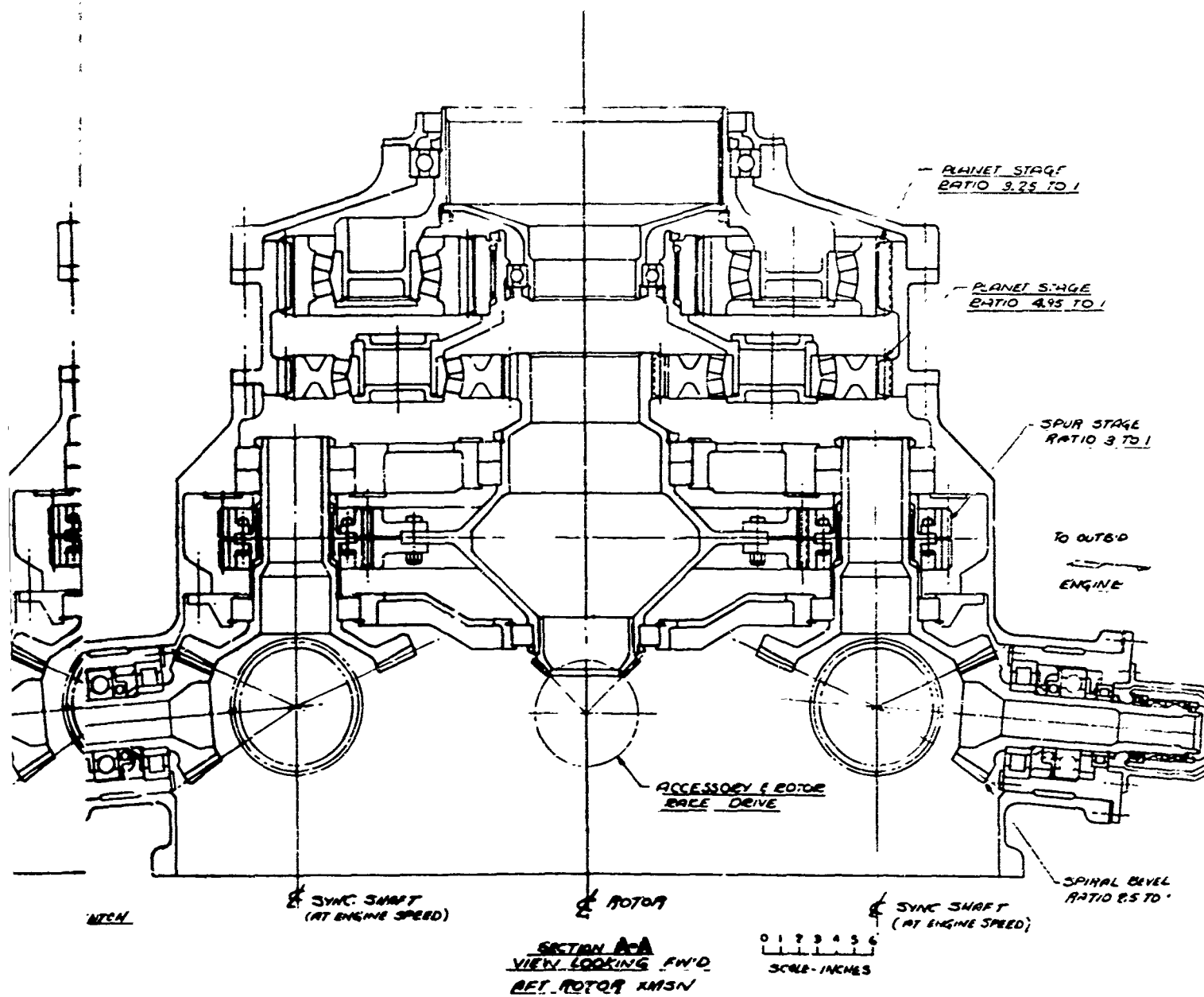


FIGURE 49. AFT ROTOR TRANSMISSION - CONFIGURATION III - SK 14243.



SK14243

**B**

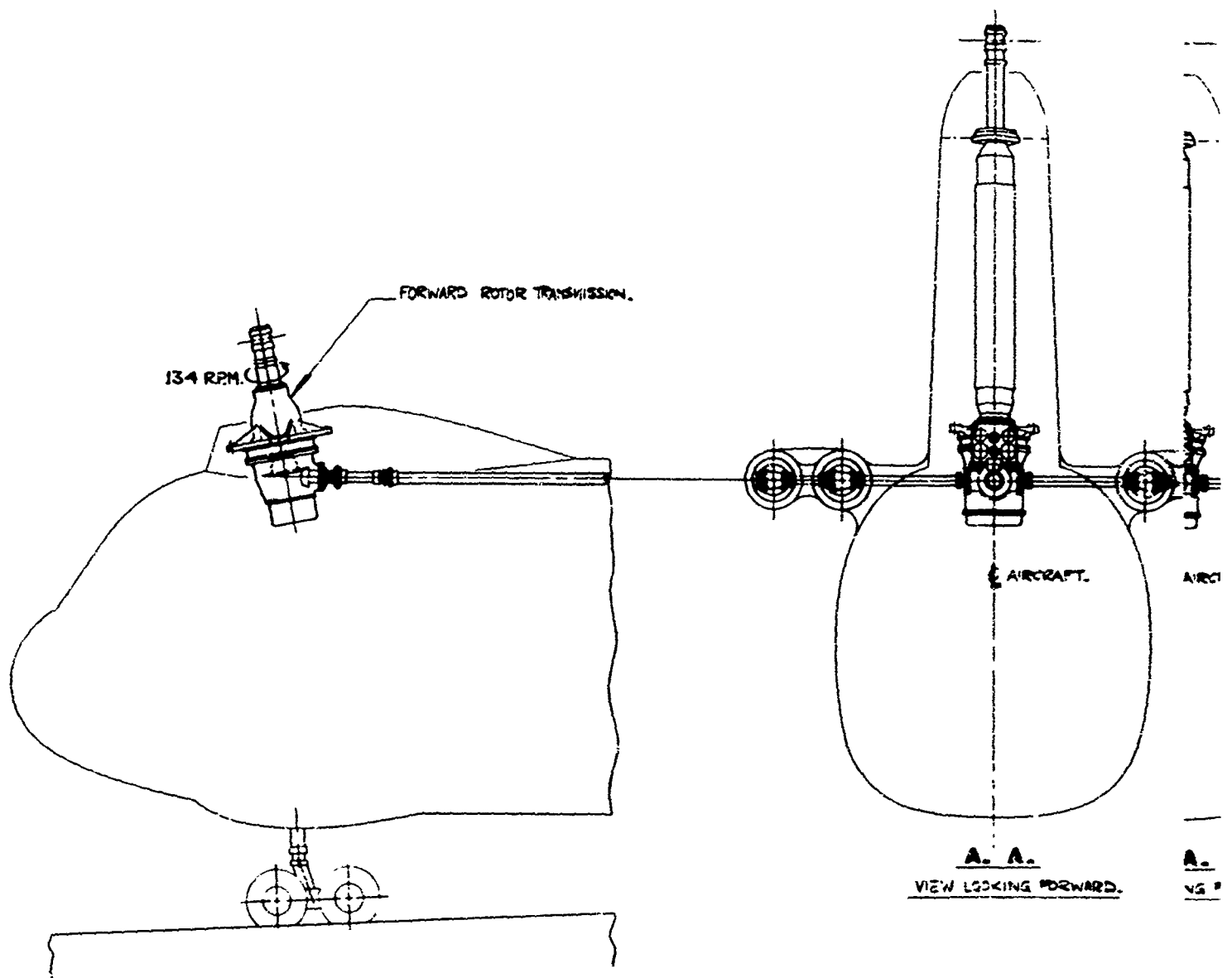
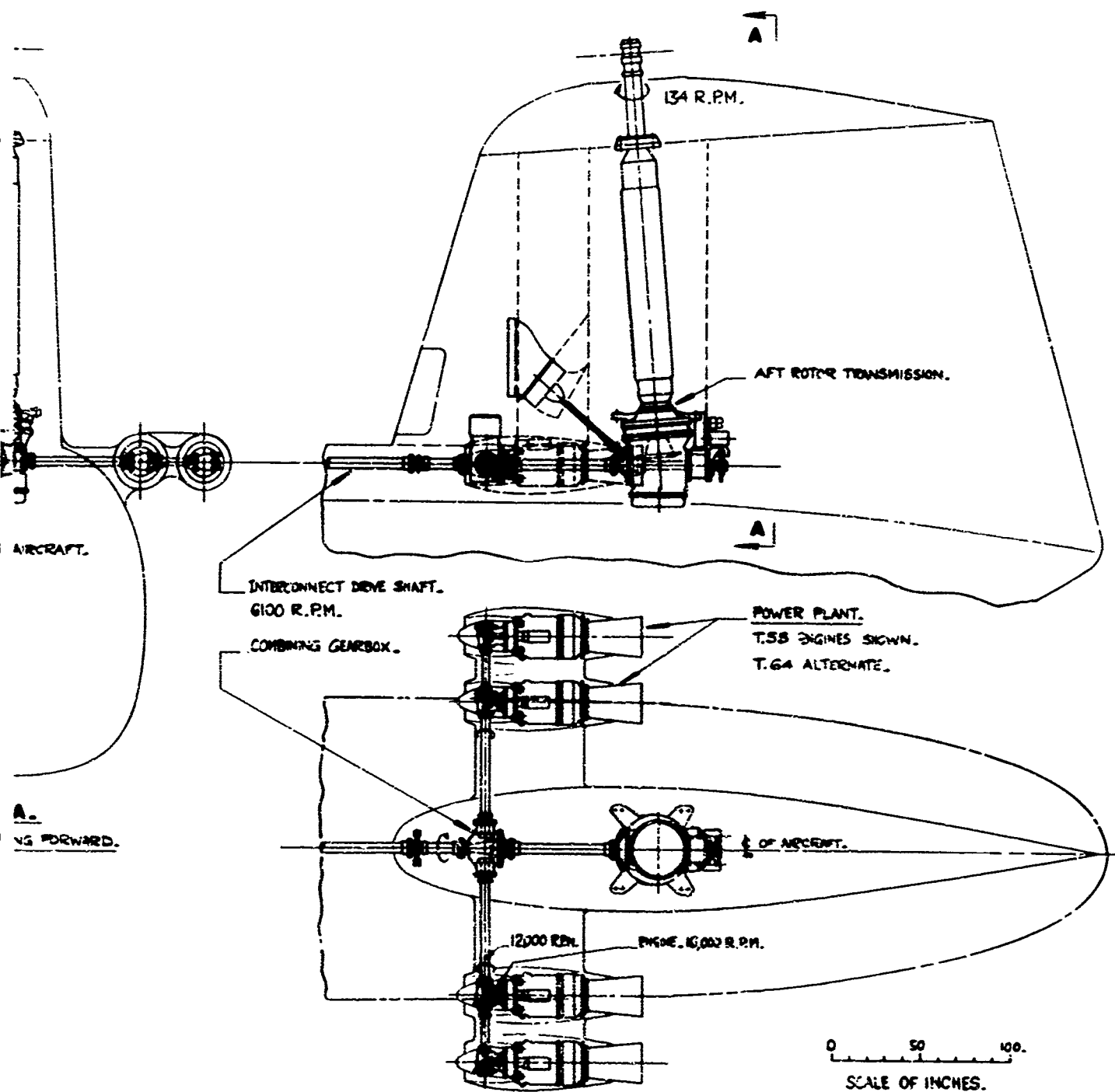


FIGURE 50. STUDY CONFIGURATION IV - SK 14245.



**B**

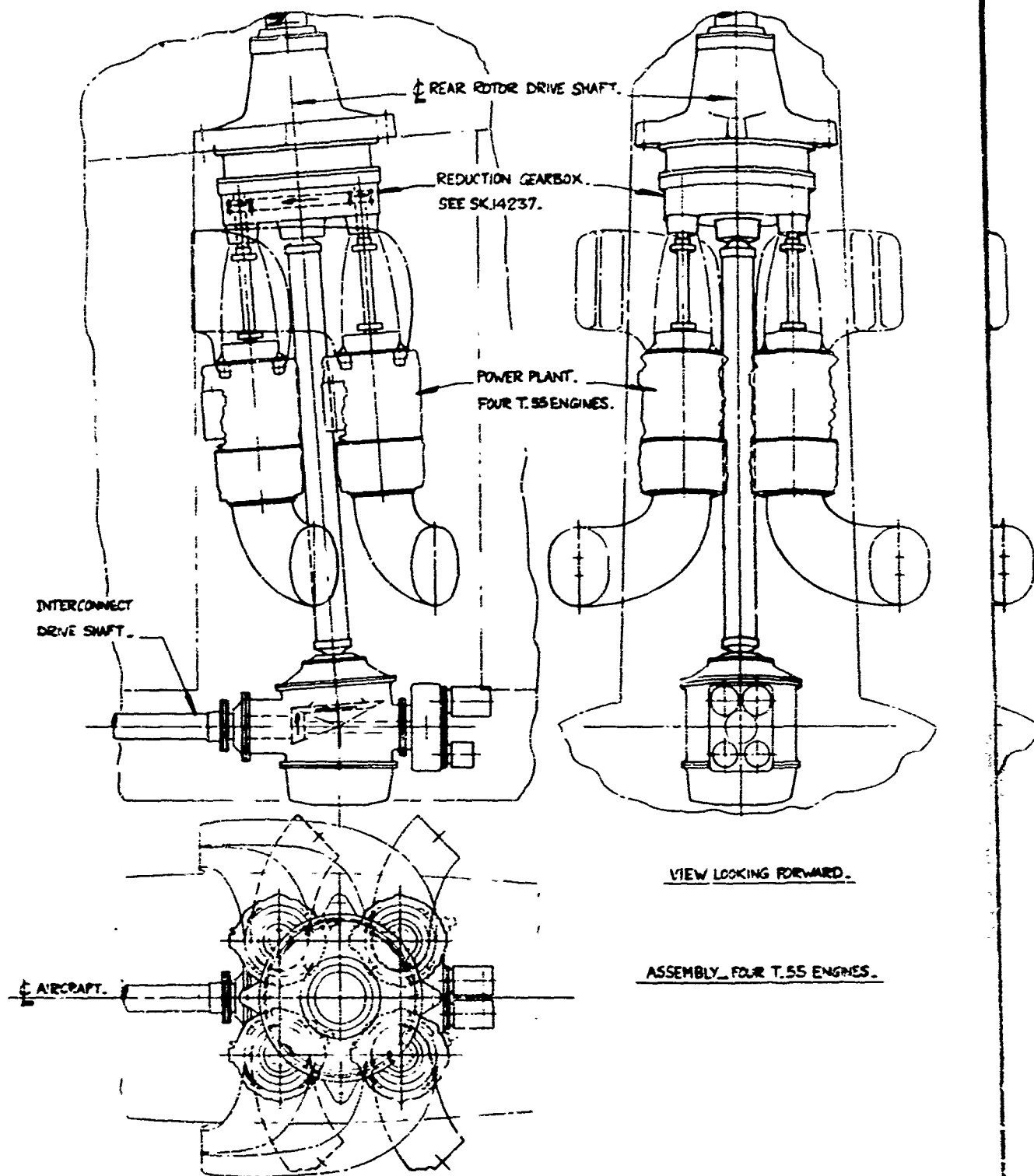
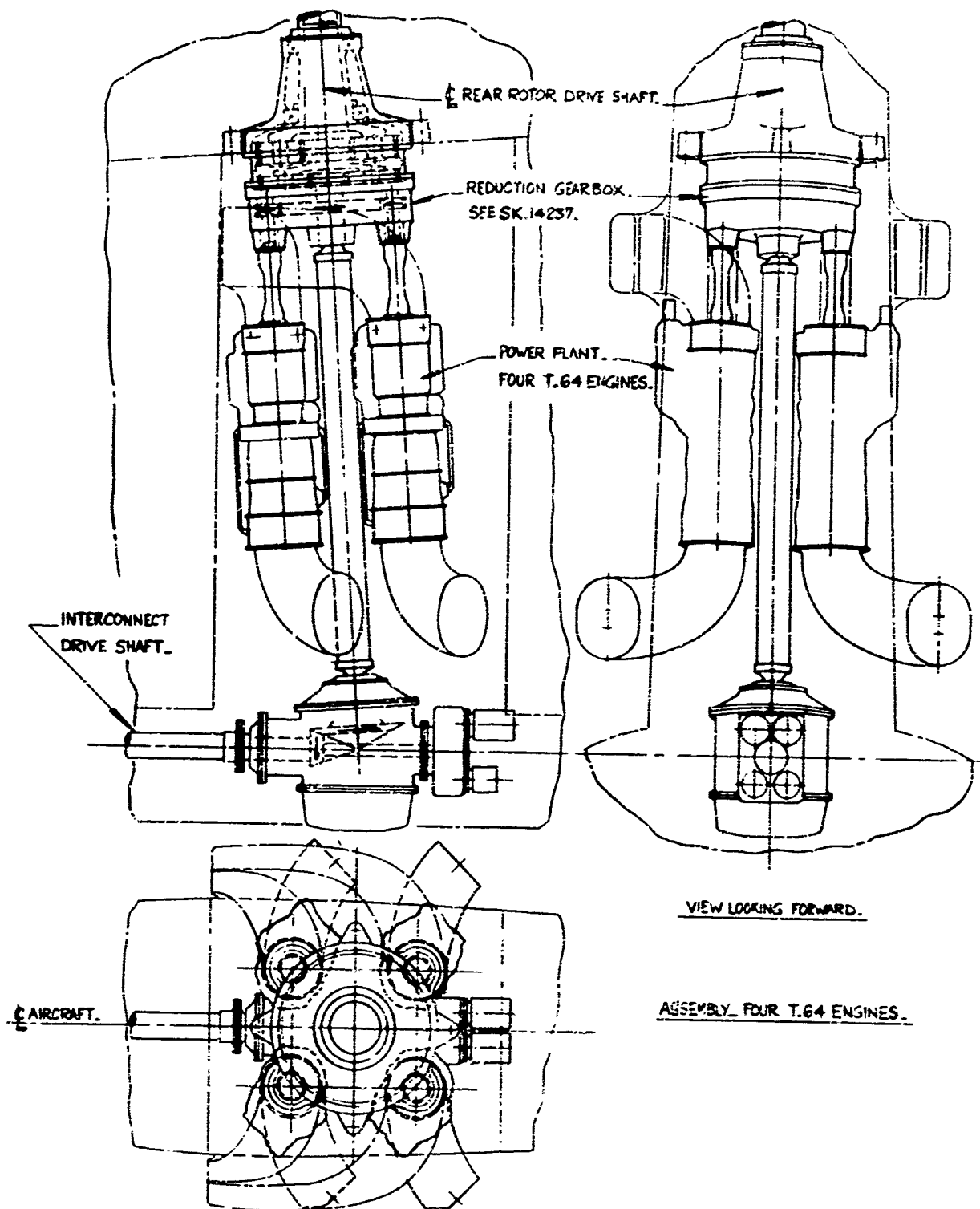


FIGURE 51. DRIVE SYSTEM - CONFIGURATION V - SK 14236.



**B**

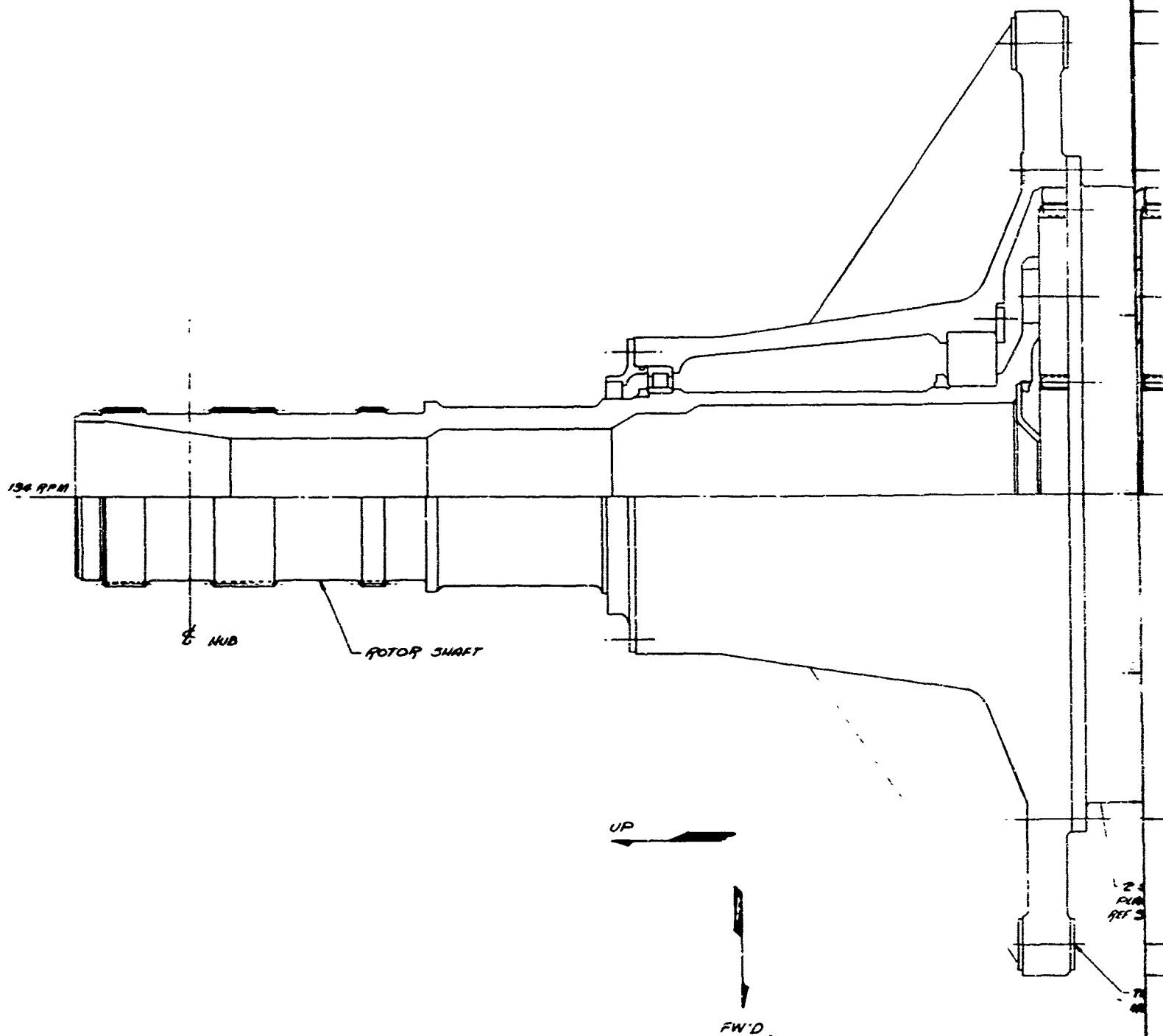
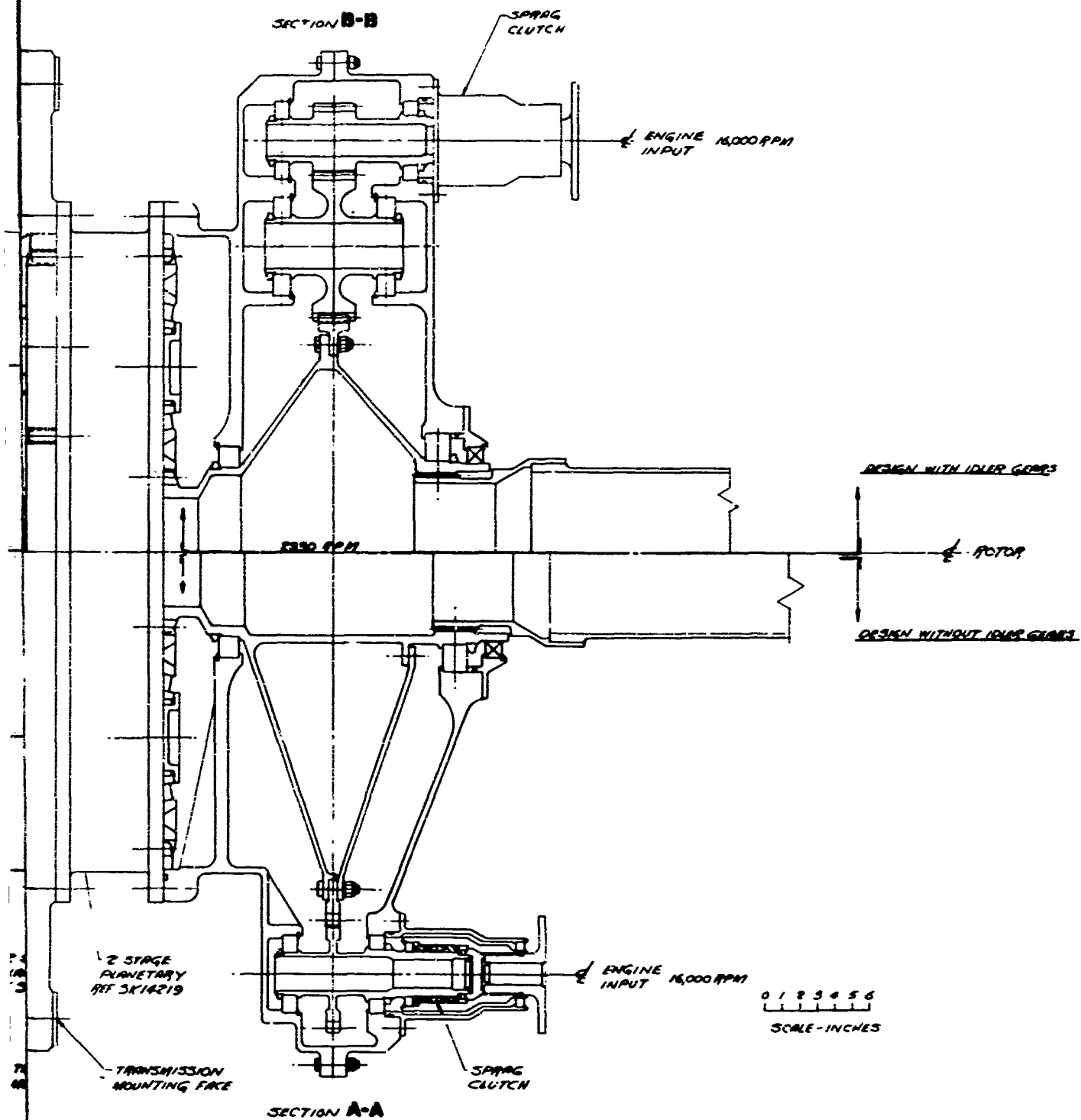


FIGURE 52. COMBINING TRANSMISSION - CONFIGURATION V -  
SK 14237. (Sheet 1 of 2)



**B**

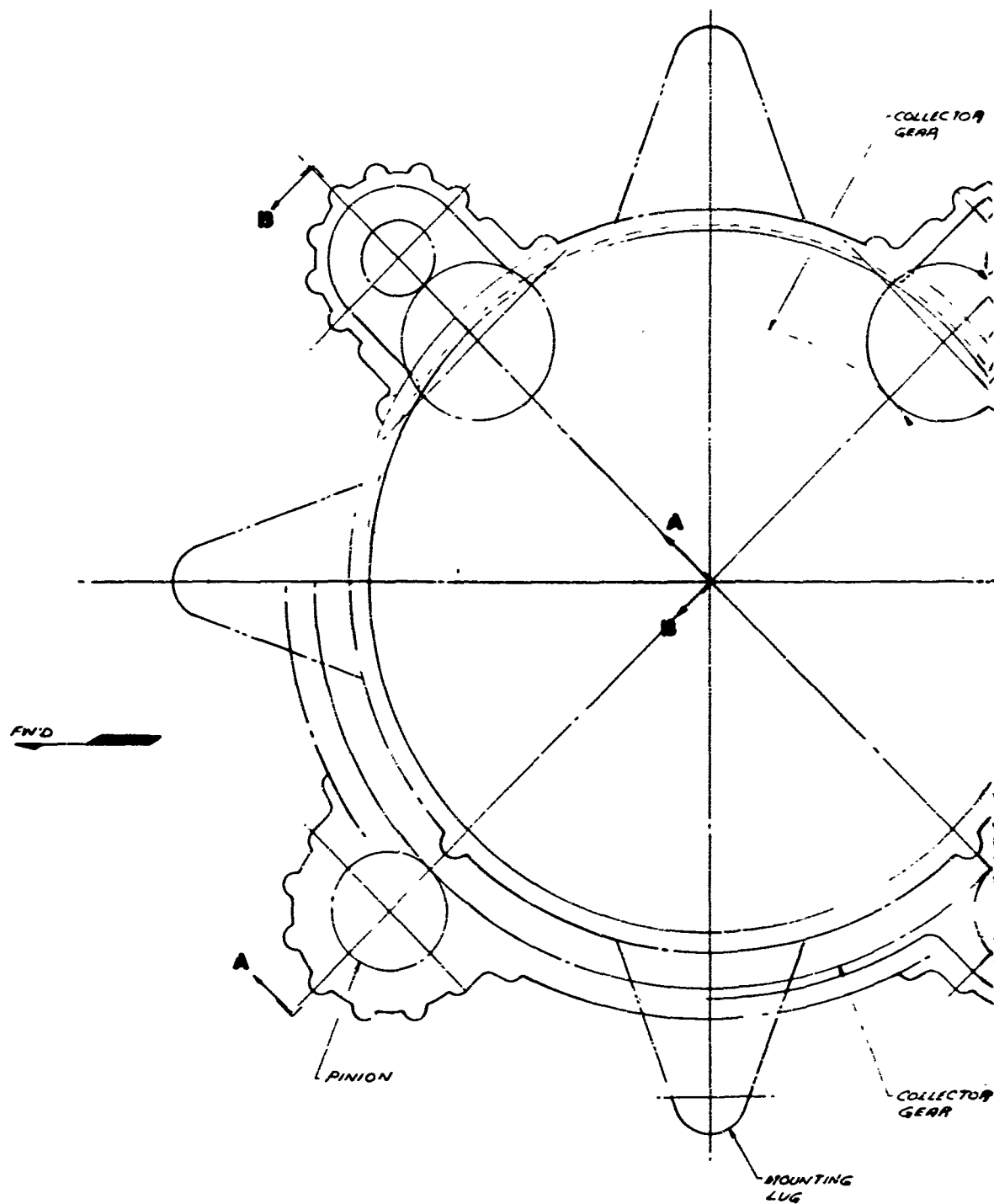
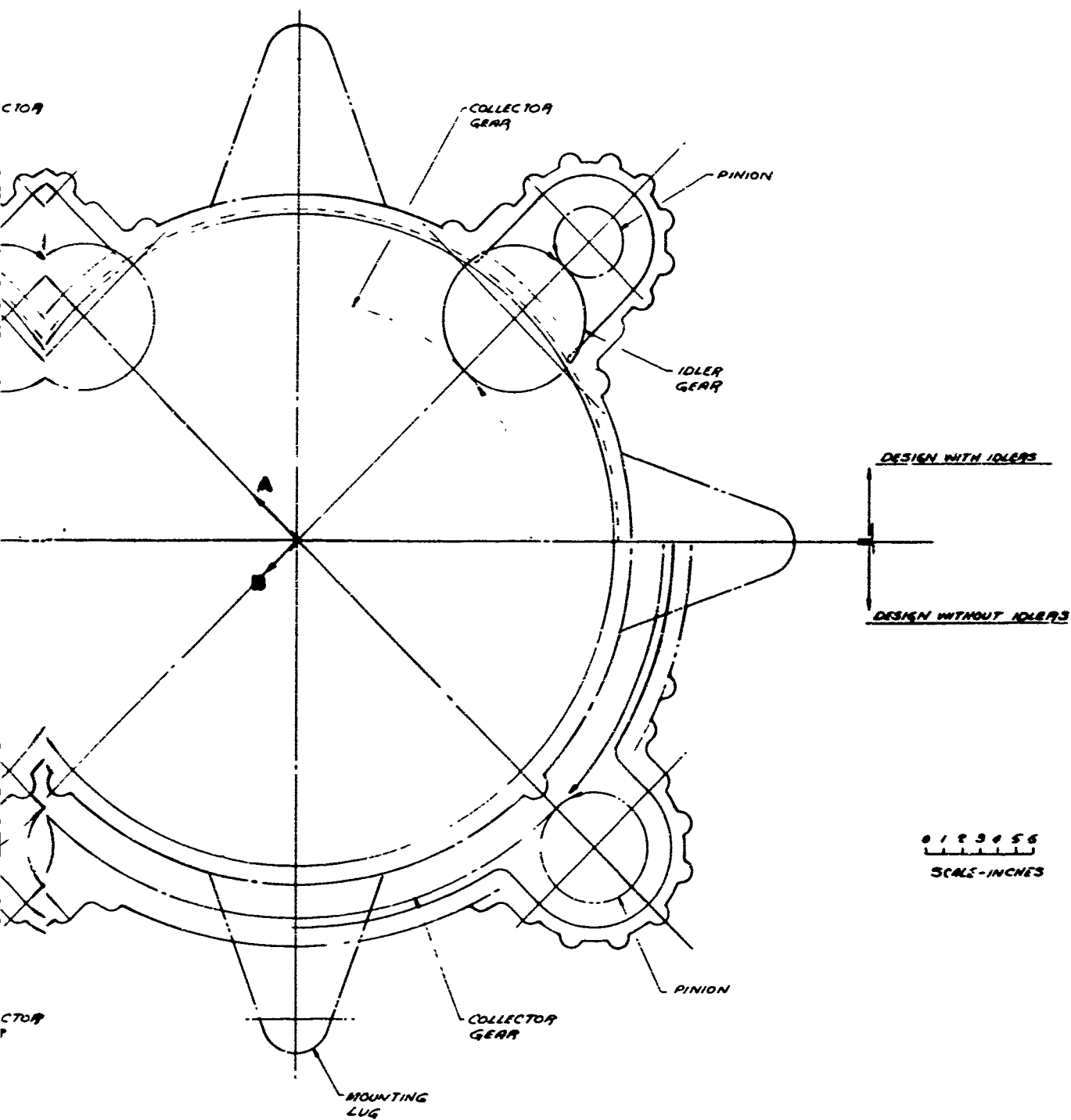


FIGURE 52. COMBINING TRANSMISSION - CONFIGURATION V - SK 14237. (Sheet 2 of 2)



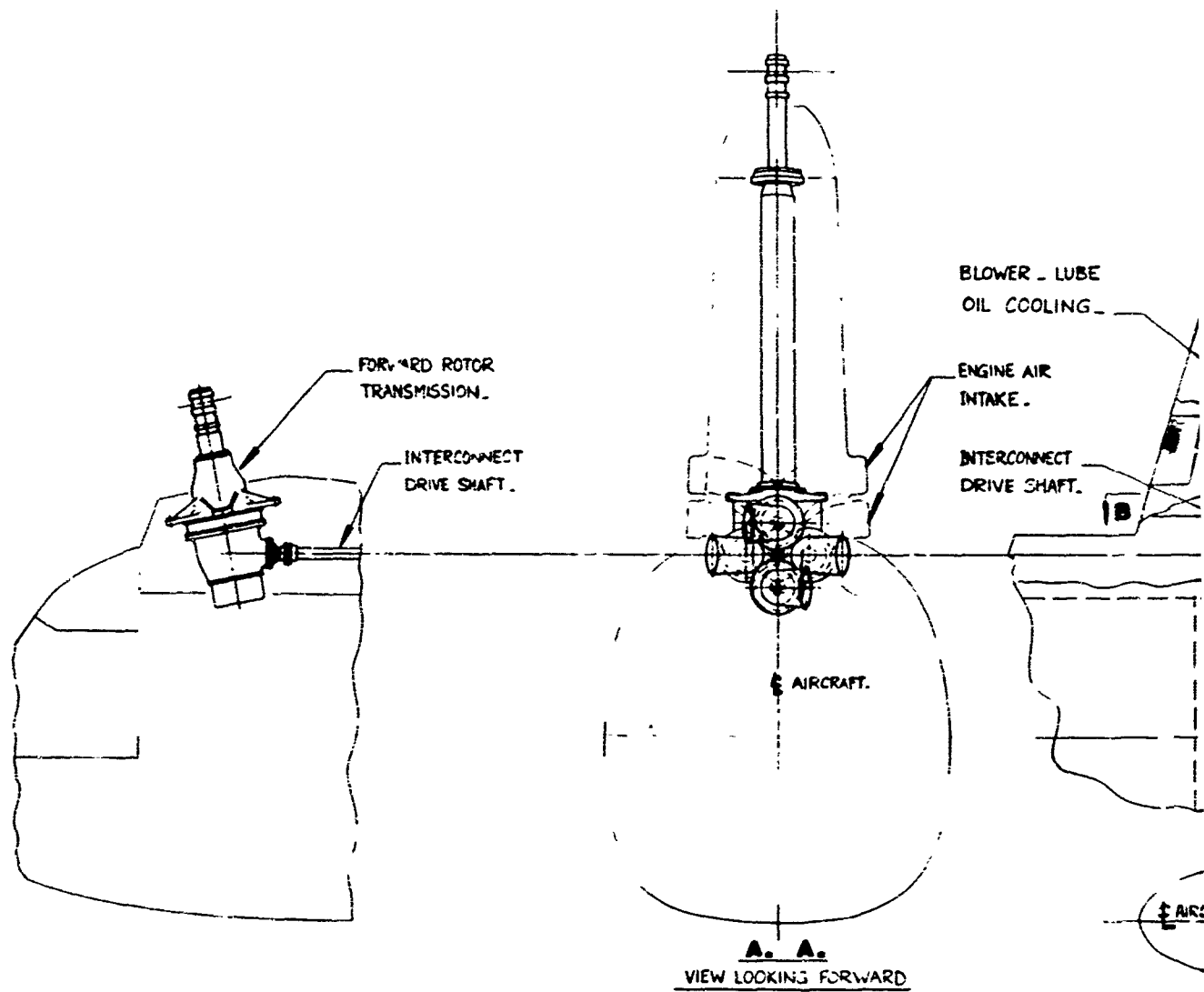
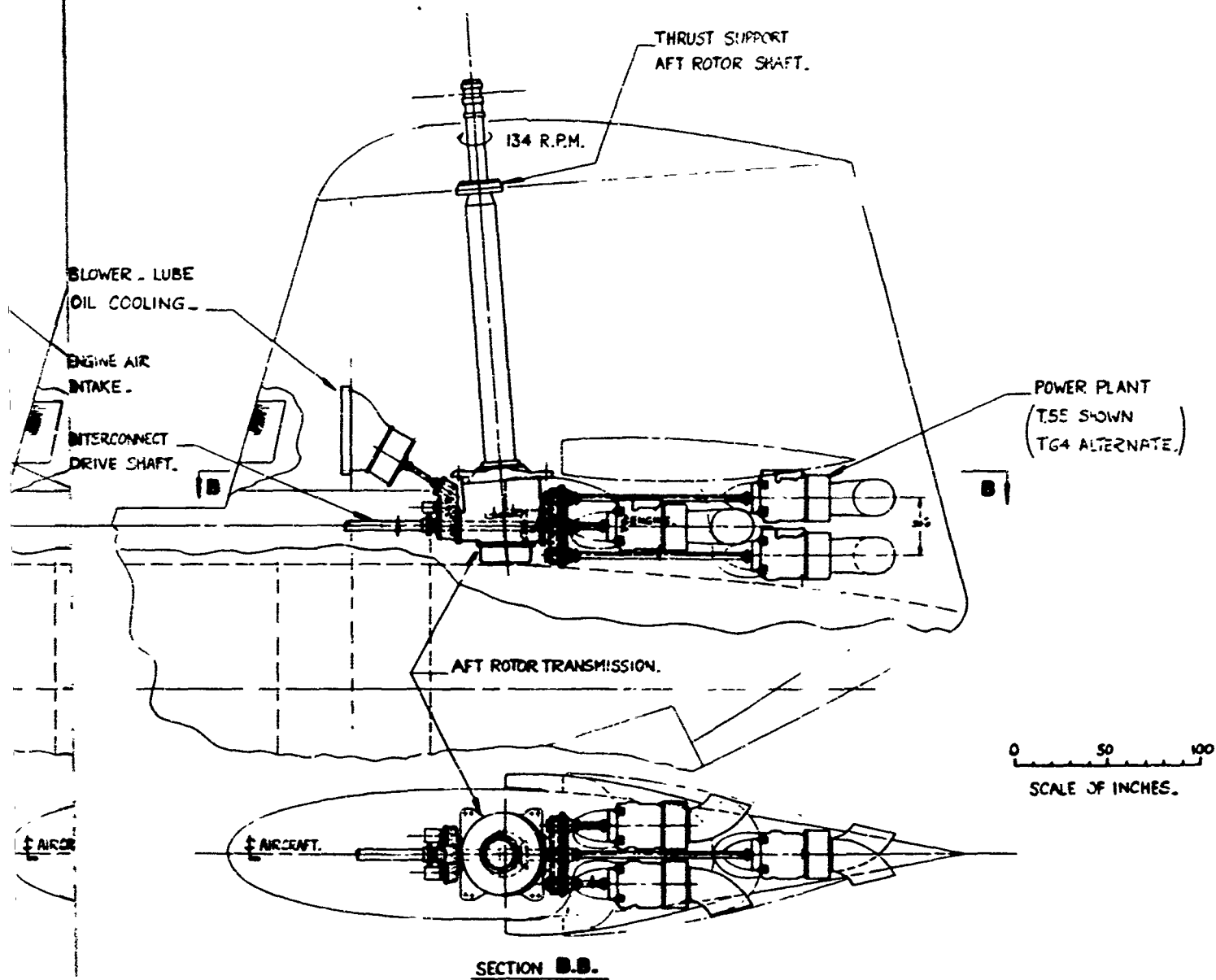


FIGURE 53. STUDY CONFIGURATION VI - SK 14244.



**B**

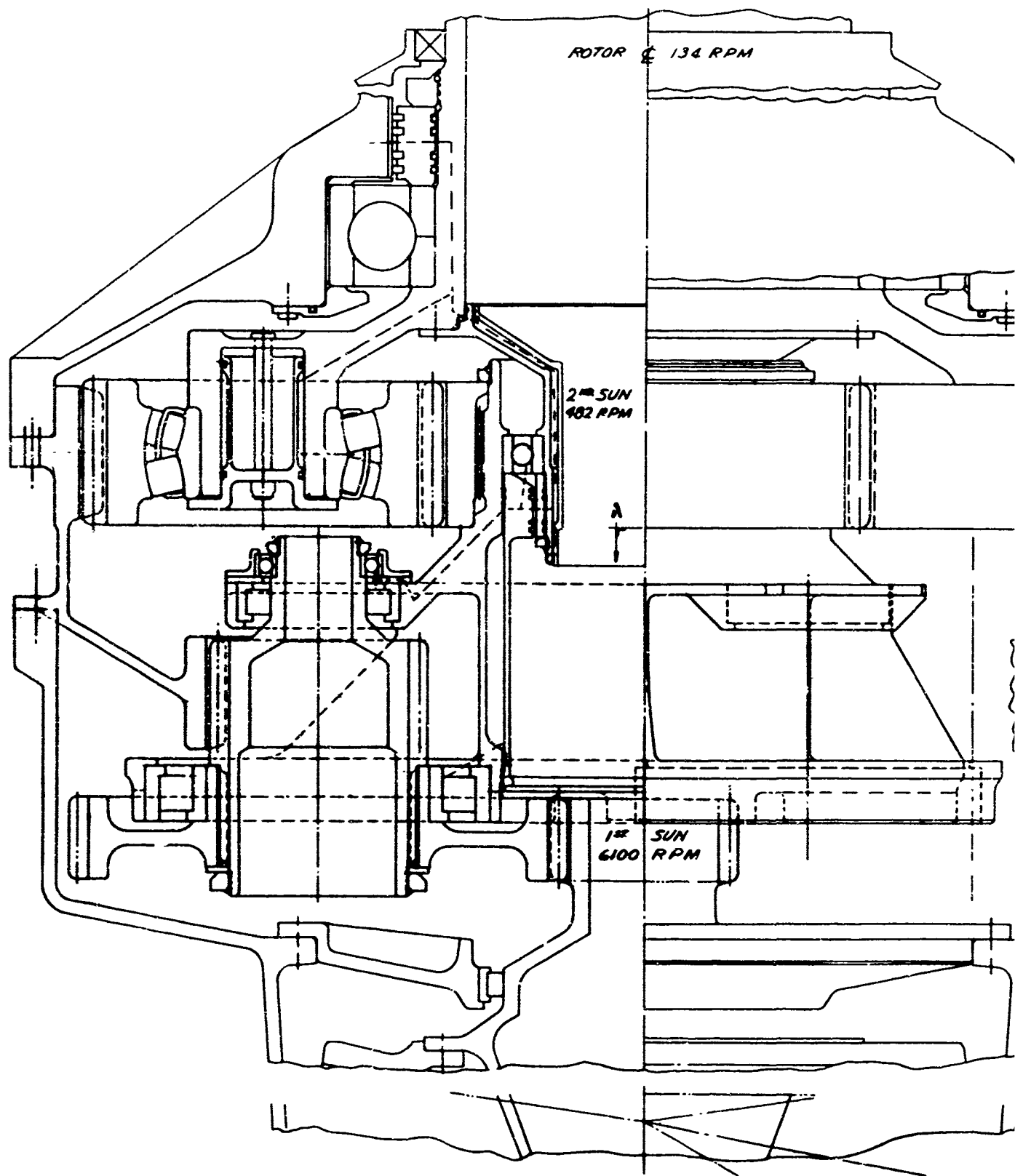
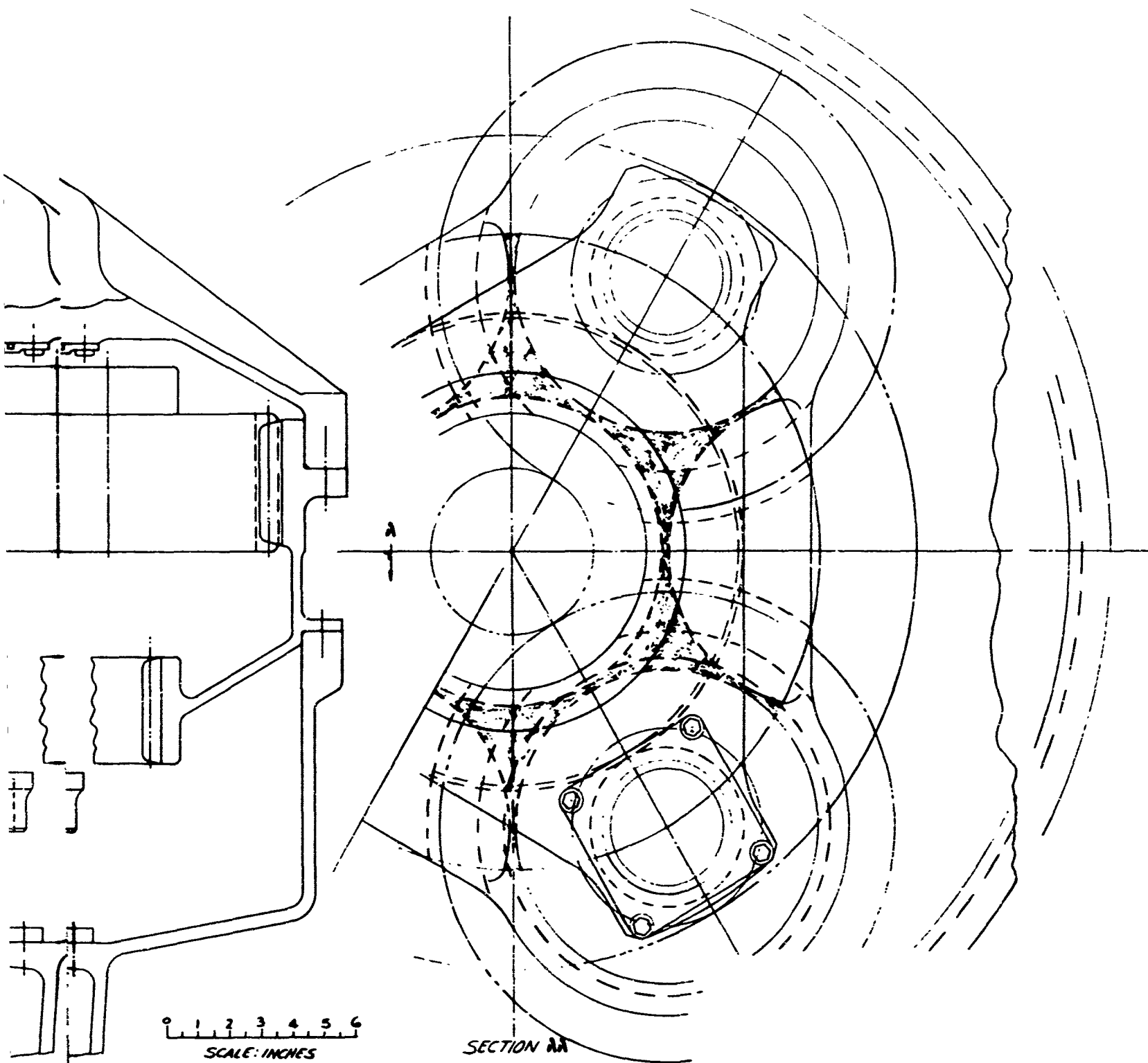


FIGURE 54. COMPOUND REDUCER STUDY - SK 14232.



**B**

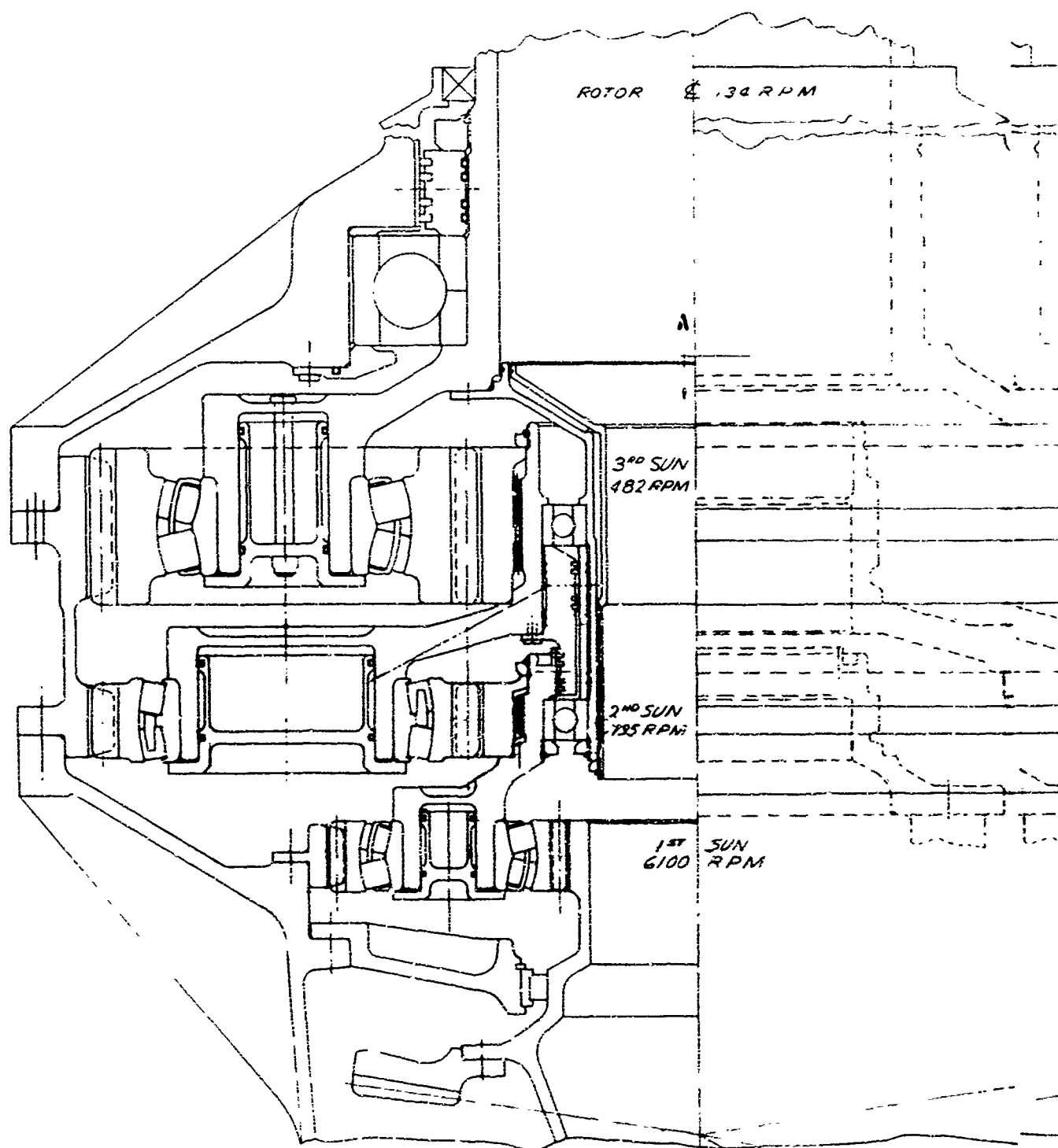
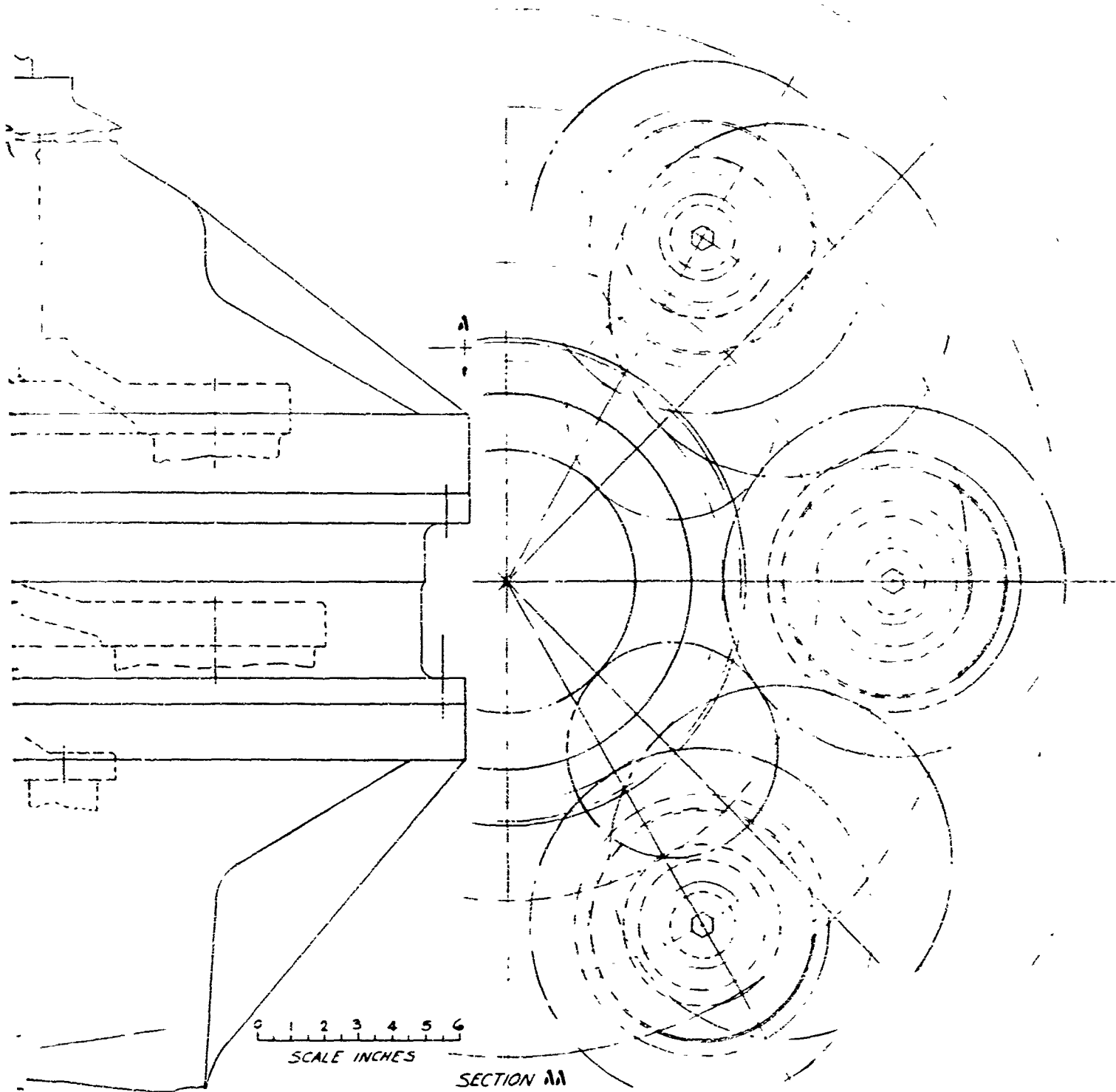


FIGURE 55. THREE-STAGE REDUCER STUDY - SK 14231.



**B**

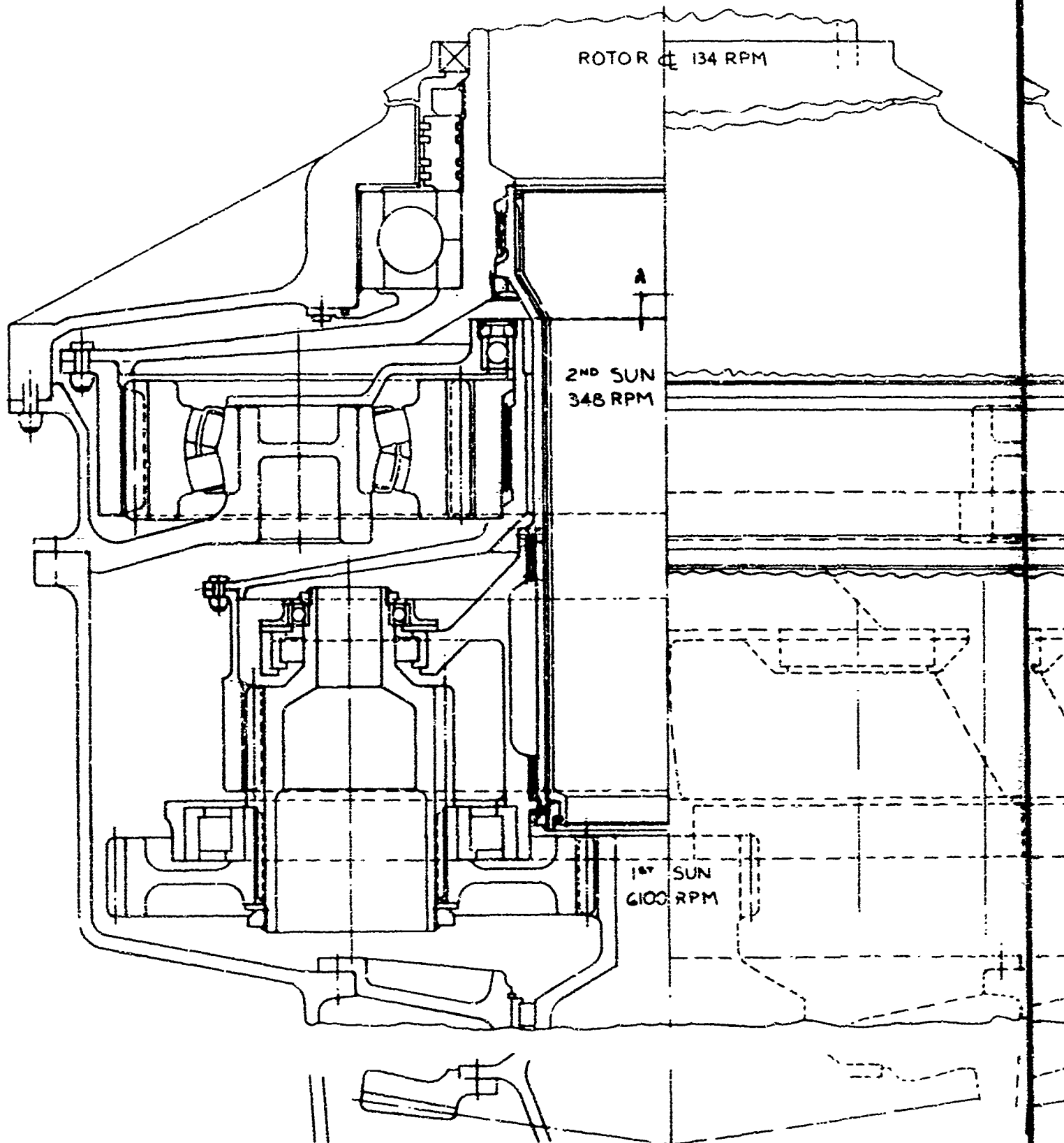
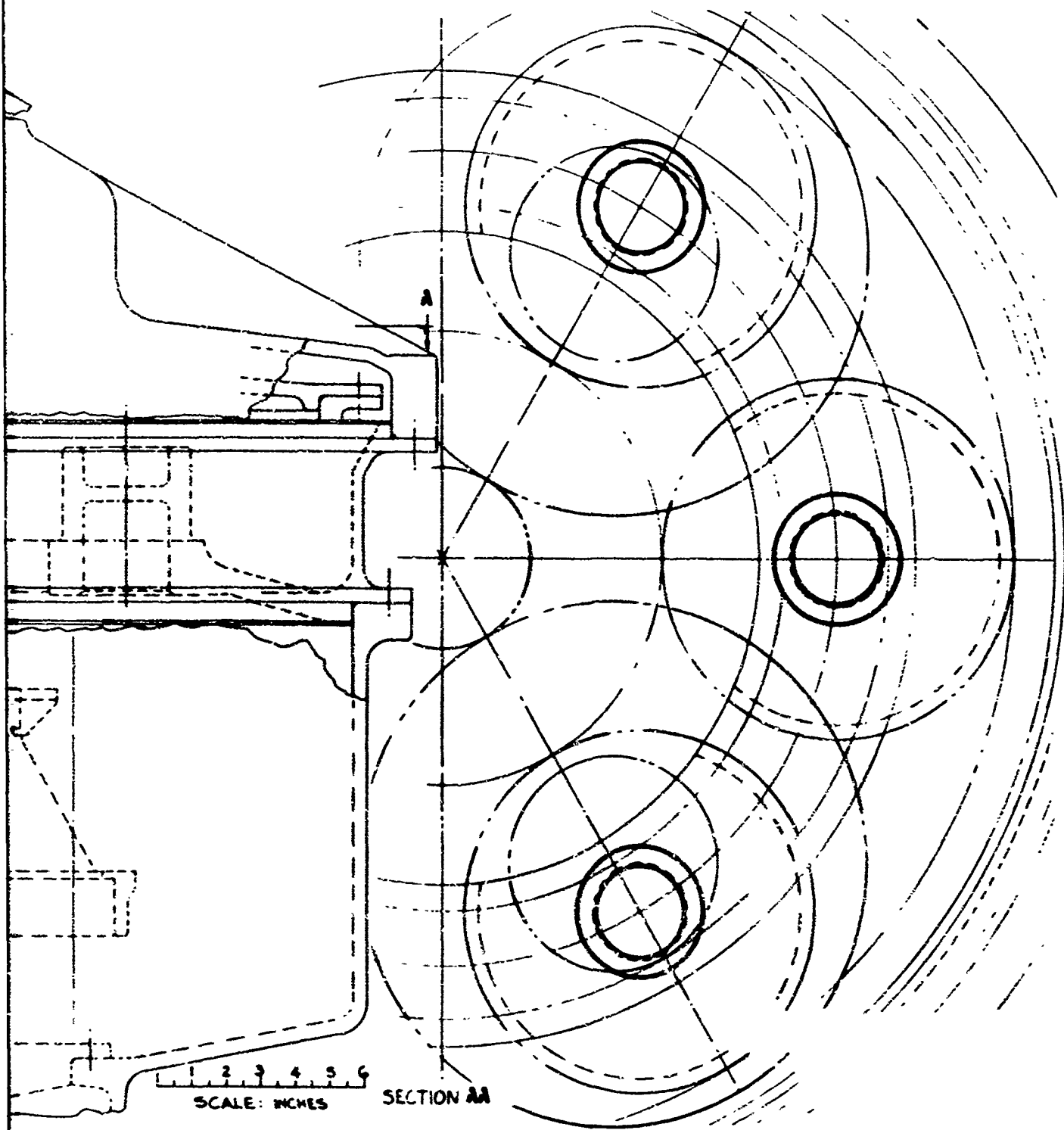


FIGURE 56. SPLIT-TORQUE REDUCER STUDY - SK 14230.



**B**

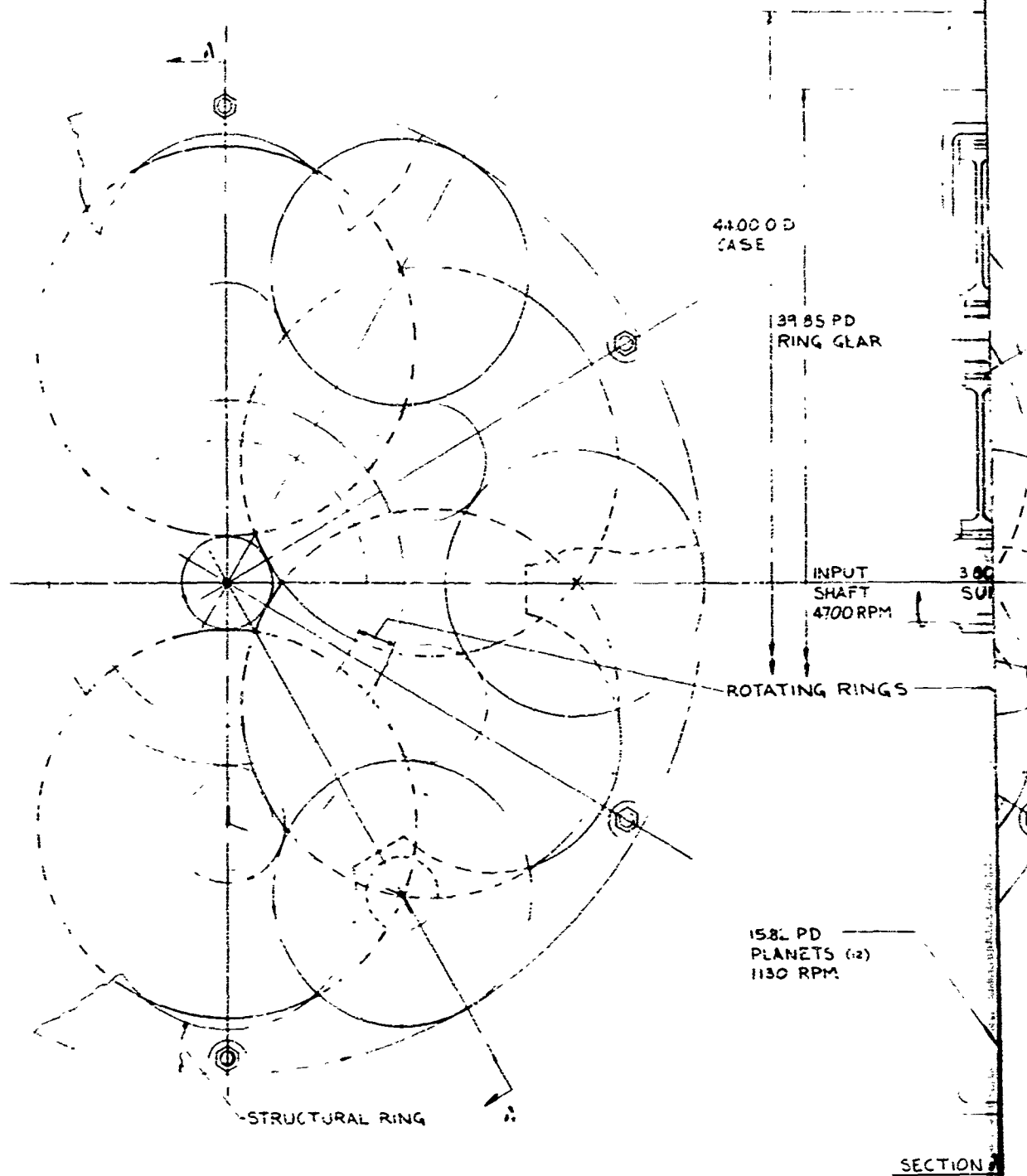
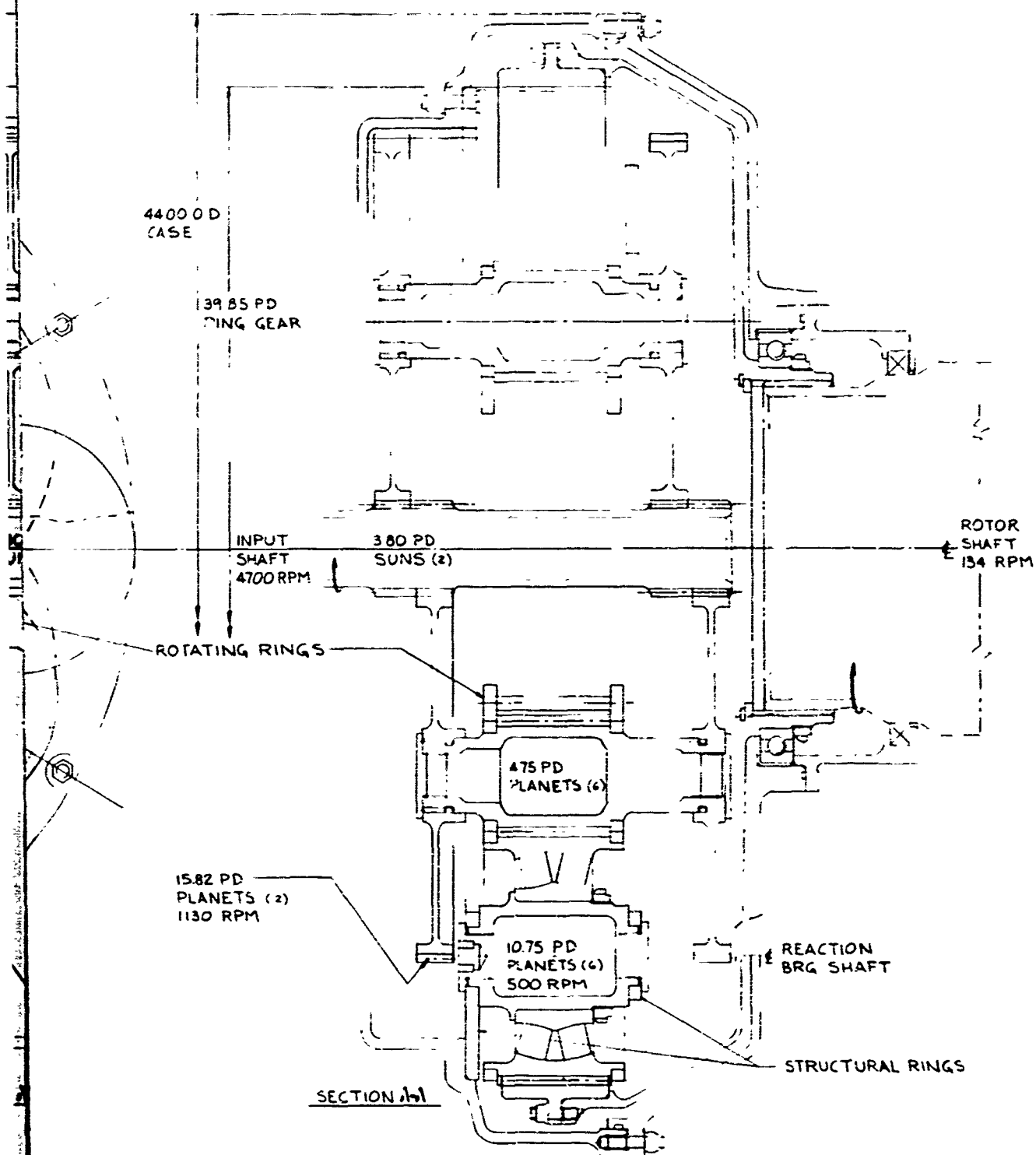


FIGURE 57. ROLLER GEAR REDUCER STUDY - SK 14233.



- SK 14233.

**B**

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2. Harris, T., and Broschard J., Analysis of an Improved Planetary Gear Transmission Bearing, SKF Industries, Inc.
3. Morrison, L.D., et al, "The Effect of Material Variables on the Fatigue Life of AISI 52100 Steel Ball Bearings", ASLE Transactions, Volume 5, No. 2, November 1962.
4. McIntyre, W.L., How to Reduce Gear Vibration Failure, paper given before AGMA, February 1964.
5. Palmgren, A., Ball and Roller Bearing Engineering, SKF Industries, Inc.
6. Stalter, J.L., Wind Tunnel Test of a 1/8-Scale Model of the YHC-1B Helicopter, Engineering Research Report No. 344, University of Wichita, Wichita, Kansas, August 1959.
7. The Vertol Division of Boeing, Heavy Lift Helicopter Study, Report R-283, prepared for USAAVLABS, Fort Eustis, Virginia, June 1962.
8. Gleason Gear Works, "Improved Method for Estimating Fatigue Life of Bevel and Hypoid Gears", SAE Quarterly Transactions, Vol. 6, No. 2, April 1952.

## APPENDIX I

### EQUATIONS FOR EQUIVALENT NUMBER OF MESHES ( $M_e$ ) FOR POWER LOSS COMPARISONS

$m_a$  = actual number of meshes  
 $R$  = total ratio

#### For Single-Stage Epicyclic Types:

Simple planetary:  $m_a = 2$ ;  $M_e = m_a \left(1 - \frac{1}{R}\right)$

Compound planetary:  $m_a = 2$ ;  $M_e = m_a \left(1 - \frac{1}{R}\right)$

Split-torque planetary:  $m_a = 4$ ;  $M_e = m_a \left(1 - \frac{1}{R} - \frac{D_3}{2RD_1}\right)$

Where

$D_1$  = diameter first sun gear  
 $D_3$  = diameter first ring gear

#### For Study Arrangements (Figure 18):

Split-torque compound:  $m_a = 4$ ;  $M_e = m_a \left(1 - \frac{1}{R} - \frac{D_4D_2}{2RD_3D_1}\right)$

Where

$D_1$  = diameter of first sun gear  
 $D_2$  = diameter of gear of compound planetary  
 $D_3$  = diameter of pinion of compound planetary  
 $D_4$  = diameter of first ring gear

Compound-simple planetary:  $m_a = 4$ ;  $M_e = m_a \left(1 - \frac{1}{2R_C} - \frac{1}{2R_S}\right)$

Where

$R_C$  = compound planetary ratio  
 $R_S$  = simple planetary ratio

Three-stage planetary:  $ma = 6$ ;  $Me = ma \left( 1 - \frac{1}{3R_1} - \frac{1}{3R_2} - \frac{1}{3R_3} \right)$

Where

$R_1$  = first-stage ratio  
 $R_2$  = second-stage ratio  
 $R_3$  = third-stage ratio

Epicyclic Roller Gear:

Odd number planet rows:  $Me = ma \left( 1 - \frac{1}{R} \right)$

Even number planet rows:  $Me = ma \left( 1 + \frac{1}{R} \right)$

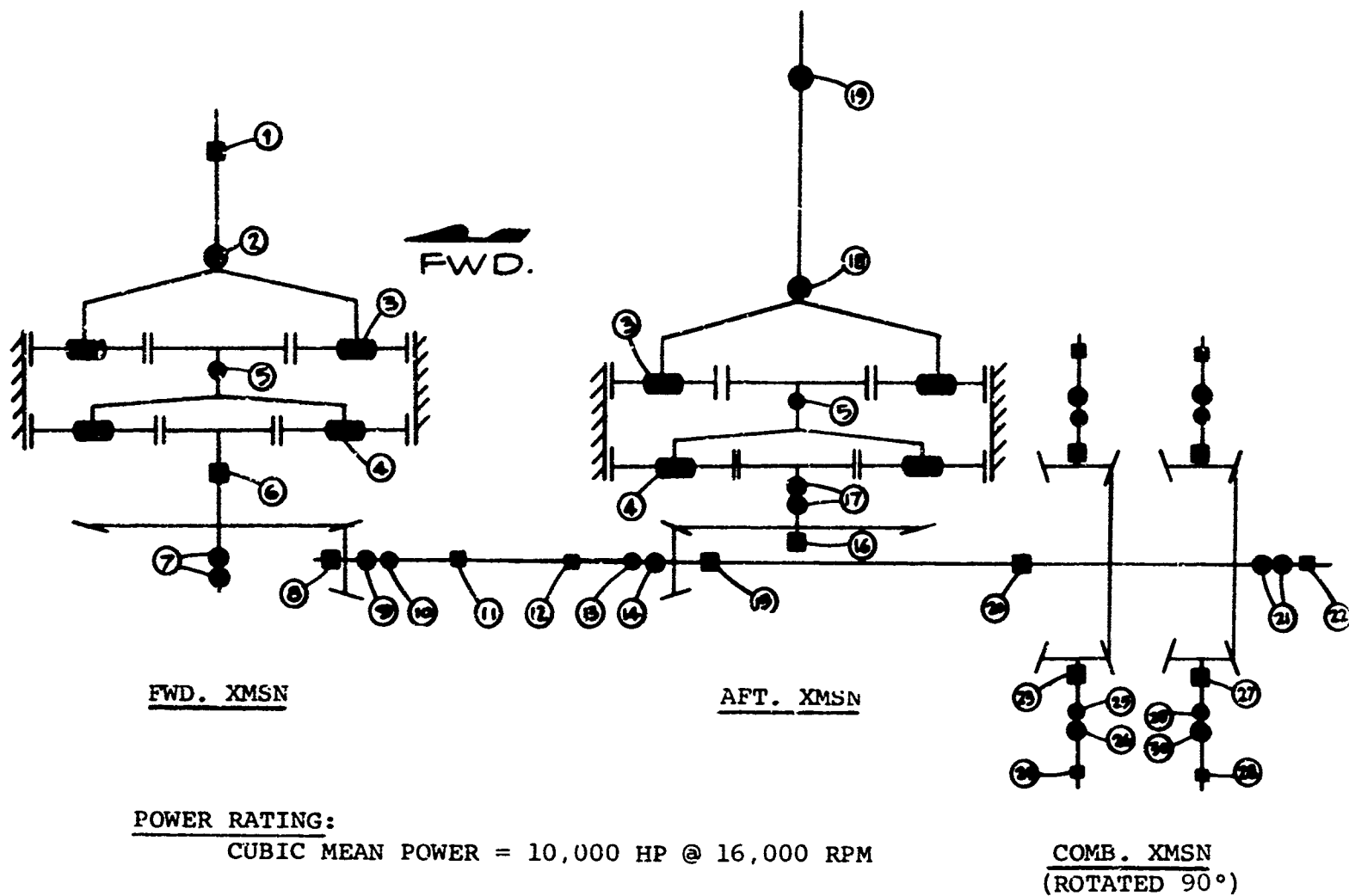
Where

$R$  = total ratio

Roller gear (Figure 57):  $ma = 3 = Me$

## APPENDIX II

### SUMMARY OF BEARING LOADS AND GEAR STRESS LEVELS FOR CONFIGURATION I



#### POWER RATING:

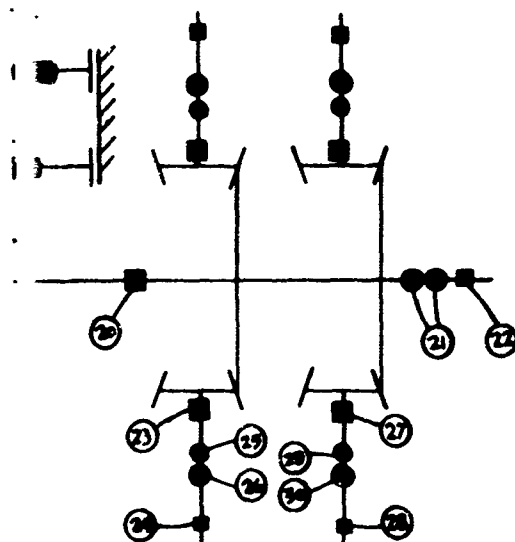
CUBIC MEAN POWER = 10,000 HP @ 16,000 RPM

COMB. XMSN  
(ROTATED 90°)

- NOTES:**
1. All B10 lives shown are based on four-engine operation with the exception of bearings (20) & (22) which are based on a two-engine-out condition.
  2. All bearing capacities were increased by a 70% growth factor.

FIGURE 58. BEARING SUMMA

3 LEVELS



COMB. XMSN  
(ROTATED 90°)

engine operation  
(22) which are  
by a 70%

LOCATION		TYPE OF BEARING	BEARING SIZE	LOADS		R P M
				RADIAL	THRUST	
FWD. XMSN	1	Roller	300x380x38	49,000	--	134
	2	Ball	320x480x74	39,000	51,000	134
	3	Spher-Rol	95x200x67	57,000	--	443
	4	Spher-Rol	110x200x53	27,000	--	1270
	5	Ball	160x220x28	--	--	340
	6	Roller	220x270x24	9,400	--	2350
	7	Ball	130x200x33	6,100	3,000	2350
	8	Roller	90x160x30	9,000	--	6150
	9	Ball	110x200x38	--	7,000	6150
	10	Ball	105x160x26	--	1,750	6150
	11	Roller	105x160x26	5,500	--	6150
AFT. XMSN	12	Roller	105x160x26	6,000	--	6150
	13	Ball	105x160x26	--	1,950	6150
	14	Ball	110x200x33	--	7,800	6150
	15	Roller	130x200x33	9,500	--	6150
	16	Roller	150x210x28	11,500	--	2350
	17	Ball	240x309x30	5,200	3,100	2350
	18	Ball	410x490x38	5,500	--	134
	19	Ball	320x480x74	14,000	51,000	134
COMB. XMSN	20	Roller	130x200x33	6,000	--	6150
	21	Ball	130x200x33	--	4,150	6150
	22	Roller	120x180x28	11,000	--	6150
	23	Roller	75x160x37	7,100	--	16000
	24	Roller	70x125x24	2,500	--	16000
	25	Ball	70x110x20	--	690	16000
	26	Ball	70x150x35	--	2,750	16000
	27	Roller	75x160x37	7,000	--	16000
	28	Roller	70x125x24	2,500	--	16000
	29	Ball	70x110x20	--	690	16000
	30	Ball	70x150x35	--	2,750	16000

FIGURE 58. BEARING SUMMARY - HLH CONFIGURATION I.

R P M	BEARING SIZE	LOADS		R P M	D · N	CAPACITY REQUIRED	BEARING CAPACITY	B <sub>10</sub> LIFE
		RADIAL	THRUST					
34	300x380x38	49,000	--	134	40,200	97,000	121,500	2560
34	320x480x74	39,000	51,000	134	43,000	152,000	198,000	2700
443	95x200x67	57,000	--	443		163,000	165,000	1300
1270	110x200x53	27,000	--	1270		105,500	131,000	2500
340	160x220x28	--	--	340	54,300	--	--	--
2350	220x270x24	9,400	--	2350	517,000	43,750	52,750	2200
2350	130x200x33	6,100	3,000	2350	306,000	45,000	46,800	1420
6150	90x160x30	9,000	--	6150	553,000	56,000	61,000	1570
6150	110x200x38	--	7,000	6150	677,000	52,500	63,000	1970
6150	105x160x26	--	1,750	6150	646,000	13,100	28,000	11100
6150	105x160x26	5,500	--	6150	646,000	34,000	48,500	3800
6150	105x160x26	6,000	--	6150	646,000	37,000	48,500	2950
6150	105x160x26	--	1,950	6150	646,000	14,800	28,000	8130
6150	110x200x38	--	7,800	6150	677,000	59,000	63,000	1440
6150	130x200x33	9,500	--	6150	800,000	59,000	75,000	2640
2350	150x210x28	11,500	--	2350	353,000	53,750	64,000	2160
2350	240x309x30	5,200	3,100	2350	565,000	29,000	34,000	1990
134	410x490x38	5,500	--	134	55,000	11,000	40,000	47200
134	320x480x74	14,000	51,000	134	43,000	124,000	198,000	4980
6150	130x200x33	6,000	--	6150	800,000	21,000	75,000	12200
6150	130x200x33	--	4,150	6150	800,000	31,500	46,800	3700
6150	120x180x28	11,000	--	6150	738,000	39,000	60,000	7000
16000	75x160x37	7,100	--	16000	1,200,000	59,000	71,000	2190
16000	70x125x24	2,500	--	16000	1,120,000	21,000	39,000	9780
16000	70x110x20	--	690	16000	1,120,000	7,250	13,400	7620
16000	70x150x35	--	2,750	16000	1,120,000	29,000	43,000	3960
16000	75x160x37	7,000	--	16000	1,200,000	58,000	71,000	2340
16000	70x125x24	2,500	--	16000	1,120,000	21,000	39,000	9580
16000	70x110x20	--	690	16000	1,120,000	7,250	13,400	7620
16000	70x150x35	--	2,750	16000	1,120,000	29,000	43,000	3960

C

TABLE IX. SPIRAL BEVEL

LOCATION	NAME	DIAMETRAL PITCH	PITCH DIAMETER	NO. OF TEETH	PRESSURE ANGLE	SPIRAL ANGLE	PITCH ANGLE	SHAFT ANGLE	HAND OF SPIRAL	DIRECTION OF ROTATIONS	FACE WIDTH	
FWD. ROTOR XMSN	PINION	2.600	10.000	26	20	25	22°	100°30'	R.H.	CCW	3.780	3.7
	GEAR	2.600	26.154	68	20	25	78°30'	100°30'	L.H.	CW	3.780	3.7
AFT. ROTOR XMSN	PINION	2.600	10.000	26	20	25	20°	85°30'	L.H.	CW	3.780	3.7
	GEAR	2.600	26.154	68	20	25	65°30'	85°30'	R.H.	CCW	3.780	3.7
COMBINER BOX XMSN	FWD PINION	5.000	5.000	25	20	25	21°30'	97°30'	L.H.	CW	2.250	2.2
	FWD GEAR	5.000	13.000	65	20	25	76°	97°30'	R.H.	CCW	2.250	2.2
	AFT PINION	5.000	5.000	25	20	25	20°	82°30'	L.H.	CW	2.250	2.2
	AFT GEAR	5.000	13.000	65	20	25	62°30'	82°30'	R.H.	CCW	2.250	2.2

# AL BEVEL GEAR SUMMARY

Model: HLH - Configuration I

Power Rating: Four Engine -  
15,200 HP at 16,000 RPM

FACE WIDTH	R P M	TORQUE	BENDING STRESS	HERTZ STRESS	GLEASON K-FACTOR	FACE CONTACT RATIO	PITCH LINE VELOCITY
3.780	6150	93,600	34,400	196,000	1580	1.732	16,000
3.780	2350	245,000	34,600	196,000	975	1.732	16,000
3.780	6150	93,600	34,700	198,000	1580	1.732	16,000
3.780	2350	245,000	34,900	198,000	975	1.732	16,000
2.250	16000	15,000	34,000	210,000	1195	2.000	21,000
2.250	6150	39,000	34,000	210,000	740	2.000	21,000
2.250	16000	15,000	34,000	210,000	1195	2.000	21,000
2.250	6150	39,000	34,000	210,000	740	2.000	21,000

**B**

TABLE X. SPUR GUR (

LOCATION	NAME	DIAMETRAL PITCH	PITCH DIAMETER	NO. OF TEETH	FACE WIDTH	FILLET RADIUS	CIRCULAR TOOTH THICKNESS	PRESSURE ANGLE	R P M	
FWD. & AFT. PLAN. 1st STAGE	SUN	3.000	8.333	25	2.360	.164	.516	25°	2350	
	(4) PLANET	3.000	12.333	37	2.300	.199	.516	25°	1270	
	RING	3.000	33.000	99	2.030	--	.516	25°	--	
FWD. & AFT. PLAN. 2nd STAGE	SUN	3.000	13.000	39	4.340	.157	.515	25°	474	
	(6) PLANET	3.000	10.000	30	4.280	.160	.515	25°	443	
	RING	3.000	33.000	99	3.760	--	.515	25°	--	

UR GEAR SUMMARY

Model: HLH - Configuration I

Power Rating: Four Engines -  
15,200 HP at 16,000 RPM

TORQUE	BENDING STRESS	HERTZ STRESS	P V T FACTOR	SCORING INDEX	Y <sub>K</sub>	CONTACT RATIO	TOOTHLOAD PER INCH FACE WIDTH
61,250 245,000	36,700	184,000	2,050,000	452	.509	1.474	6220
90,750	35,600	184,000	2,570,000	452	.537	1.474	6375
243,000 971,000	35,000	100,000	--	---	.620	---	7220
202,000 1,210,000	39,400	184,500	615,000	340	.544	1.491	7160
155,000	41,200	184,500	538,000	340	.527	1.491	7250
512,000 3,070,000	40,000	125,500	--	---	.620	---	8280

**B**

# APPENDIX III.

TABLE XI. SUMMARY OF LUBRICATION SYSTEM ANALYSIS  
FOR CONFIGURATION IA

Unit	Location	Efficiency	Horse- power	(BTU/Min) Heat	(gpm) Flow
Forward	Bevel	99.35	7600	2090	6.35
Rotor XMSN	1st planet	99.03	7600	3120	9.50
	2nd planet	99.03	7600	3120	9.50
Rotor Shaft	--	-	--		
				8330	25.35
Aft	Bevel	99.35	7600	2090	6.35
Rotor XMSN	Comb.bevels	99.35	3800x4	4180	12.70
	1st planet	99.03	7600	2090	6.35
	2nd planet	99.03	7600	2080	6.35
	Accessory	98.00	250	210	0.64
				10660	32.39
Total Flow (gpm) = 25.35					
32.39					
57.74					

## Notes

- Horsepower total = 15,200, equally divided.
- BTU/Minute = hp x 42.4
- Flow (gpm) =  $\frac{\text{BTU}}{7.75 \times .53 \times \Delta T}$   
Where: 7.75 = lb per gallon MIL-L-7808  
.53 = specific heat MIL-L-7808  
 $\Delta T$  = temperature rise = 80°F
- The calculated 80°F temperature rise will result in an actual expected 45°F rise according to Vertol Division experience factor.

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KEY WORDS	LINK A		LINK B		LINK C	
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Transmission						
Tandem-rotor helicopter						
Planetary reduction gearing						
Drive shafting						
Multiengine control						
Supercritical-speed shafting						
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